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THE REGENERATIVE FEED SYSTEM COMPARED TO THE CONVENTIONAL CLOSED FEED SYSTEM IN A NAVAL VESSEL.

BY ABRAHAM SOLOV.*

INTRODUCTION.

This paper presents the results of an investigation made to determine the relative merits of a naval marine power plant employing the conventional closed feed system, with the turbine operating on the Rankine cycle, and a similar plant operating on the regenerative feed cycle.

The regenerative cycle has been extensively used in central station and other stationary plants, and also to some extent in

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merchant vessels,[1] particularly those built by the Marine Commission. Up to the present, however, this cycle has not been employed on any naval vessel.

The chief factors investigated in comparing the two plants was that of efficiency and economy. However, the important items of weight and space requirements, as well as ship maneuverability, maintenance and simplicity of operation were also considered.

General Considerations.

A plant employing the regenerative feed system will operate most efficiently on one particular load condition as will be shown later. Therefore, the first point that had to be decided at the outset of the study was at which speed was the performance of the two plants to be analyzed. Since a naval vessel operates mostly at cruising speed, except under battle conditions, it was felt that the investigation should be made of the power plant's performance at that speed.

To make the investigation as practical as possible, and in order to reduce to a minimum the number of assumptions that otherwise may have been necessary, particularly in the heat balance computations, it was decided to take a plant employing the conventional feed system that is already in actual operation on a naval vessel and compare it with a proposed plant employing the regenerative feed heating cycle.

Now, the existing plant chosen may not be of the most efficient type and may even contain some objectionable features, nevertheless, the proposed regenerative plant is patterned directly after it, and is almost identical to it except for those features which are peculiar to the regenerative feed system. In this way the two plants are placed on a comparative basis and any advantages that one plant may have over the other can be directly attributed to the regenerative feed heating feature, or to the absence of it, and not to any other improvements in plant design.

At this point it might be well to point out that some of the data used in this study would be considered as restricted from general use if identified with any particular vessel. For this

reason such information as the name, size, or type of vessel, as well as some of the sources from which that data was obtained, will have to be withheld.

The Conventional Plant.

The vessel chosen for this investigation has a cruising speed of 15 knots, requiring 11,000 shaft horsepower to propel it at this speed. The ship is driven by four turbines operating through double reduction gears. The turbines have cruising elements built into them in order to save weight and space. The heat balance for this plant is shown in Figure 1. The gland jet condensers employed use a portion of the condensate from the main and dynamo condensers to condense the gland steam. The heater shown just ahead of the deaerator in Figure 1 is an integral part of the deaerator.

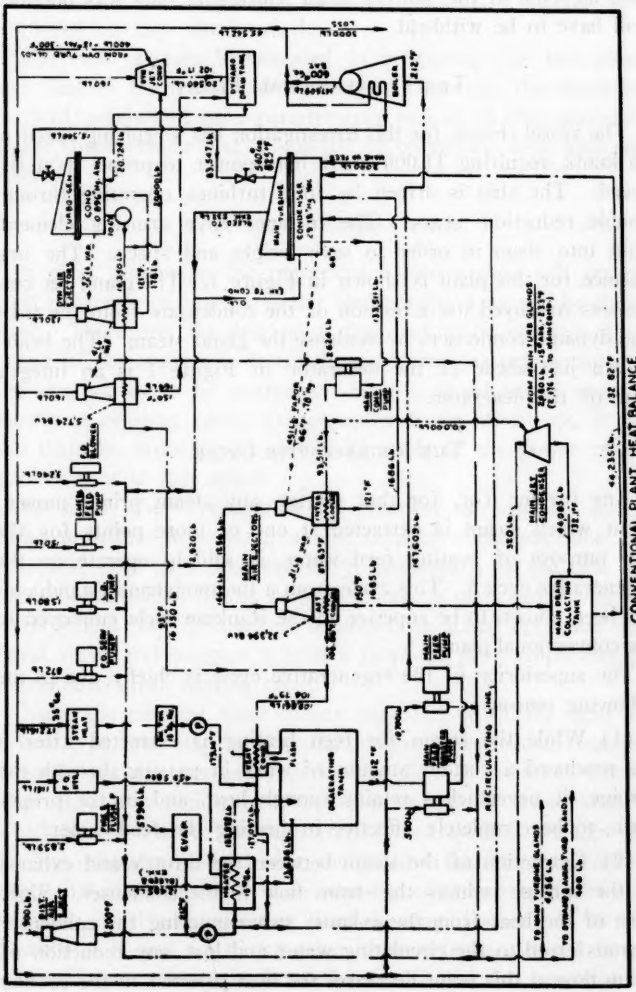
The Regenerative Cycle.

Any turbine (or, for that matter, any steam prime mover) from which steam is extracted at one or more points for the sole purpose of heating feed water, is said to operate on the regenerative cycle [2]. This cycle from a thermodynamic standpoint has been shown to be superior to the Rankine cycle employed in the conventional plant [3, 4].

The superiority of the regenerative cycle is chiefly due to the following reasons:

(1) While the steam for feed heating is extracted after it has produced a certain amount of work in passing through the turbine, it nevertheless retains enough heat, and in the proper form, to be completely effective in heating the feed water.

(2) Extraction of the steam between the throttle and exhaust of the turbine reduces the steam flow to the condenser. Since most of the heat from the exhaust steam entering the condenser is transferred to the circulating water and lost, any reduction of steam flow at this point decreases the heat rejected to the cooling water, and thereby represents a saving which is turned back to the system.



CONVENTIONAL PLANT HEAT BALANCE
Fig. 1

The regenerative cycle is particularly effective at high pressures, and as steam pressures are increased and the size of the units are also increased, application of the regenerative cycle assumes greater importance. This is due largely to two reasons: First, the thermal efficiency of the unit is increased as the pressure becomes higher, provided the turbine is of sufficiently large capacity. Second, the size and thus the cost and weight of the turbine exhaust shell, the condenser, and other related parts, become less when the regenerative cycle is substituted for the Rankine cycle. In general, this saving in large units more so than in smaller units, will usually be greater than the added expense for the regenerative feedwater heaters, and the extra piping and valves required [5].

Losses in a Regenerative Cycle.

Each unit of mechanical energy delivered by a turbine from steam that is condensed in a regenerative heater is generally produced with a very high thermal efficiency, because none of the energy of this steam is thrown away to the exhaust. The efficiency, however, is not 100 per cent because of the following losses: [6]

(1) The thermodynamic loss due to the use of a few heaters, instead of an infinite number of heaters, and due to the pressure drop between turbine and heaters.

(2) The thermodynamic loss due to the difference in temperature between steam and water in surface heaters.

(3) The internal losses of the turbine up to the point of extraction, due to leakage, turbulence and friction of the steam in passing partially through the turbine.

(4) Losses in the steam and feedwater circuits due to fluid friction, turbulence and leakage.

(5) Mechanical losses in turbine and feed pumps.

(6) Heat dissipation losses due to transfer of heat from piping, turbine casing, pumps and heaters.

CONVENTIONAL PLANT HEAT BALANCE
Fig. 1

Number of Heaters Most Economical.

The number of heaters used and the temperature to which the feedwater is raised greatly affect the thermal performance of the regenerative plant. The optimum feed water temperature to attain and the number of heaters to install are matters that must be determined for the conditions which prevail in each individual case.

In general, each heater added in the system improves the performance of the cycle, but in a lesser amount than the preceding heater, while its cost is just as much. Therefore, due to the law of diminishing returns, there is an economic limit to the number of heaters that could be used. In Marine installations, particularly on naval vessels, the increased weight of the added heaters is another factor which limits the number of heaters to be employed.

In deciding on the number of heaters and the optimum feed water temperature consideration must be given to the relation which exists between the number of heaters, feed temperature, the throttle pressure, and the exhaust pressure, engine efficiency, and energy consumption of the turbine. Numerous tests and investigations have been made on this subject. The results that may be expected from plants operating with different numbers of heaters and at different steam pressures and temperatures may be easily predicted from curves calculated by Dr. A. DeSmaele [7] and from tests conducted by the Detroit Edison Company [8], [9], and others [10]. An analytical approach to the subject has been developed by J. Kenneth Salisbury [11].

Most of the investigations and studies mentioned were made for rather large units (up to 50,000 Kw.). The results may, nevertheless, be easily applied to smaller units. Formulas and curves based on these tests, which may be applied to any size unit or plant, may be found in any modern text [12] or handbook [13], [14].

In modern practice it is usual to employ from 2 to 5 feed heaters. The larger number is more profitably employed in stationary plants with large high pressure units [15]. In marine plants,

due to space and weight limitations, and the need for a minimum of piping, three heaters is the number often used.

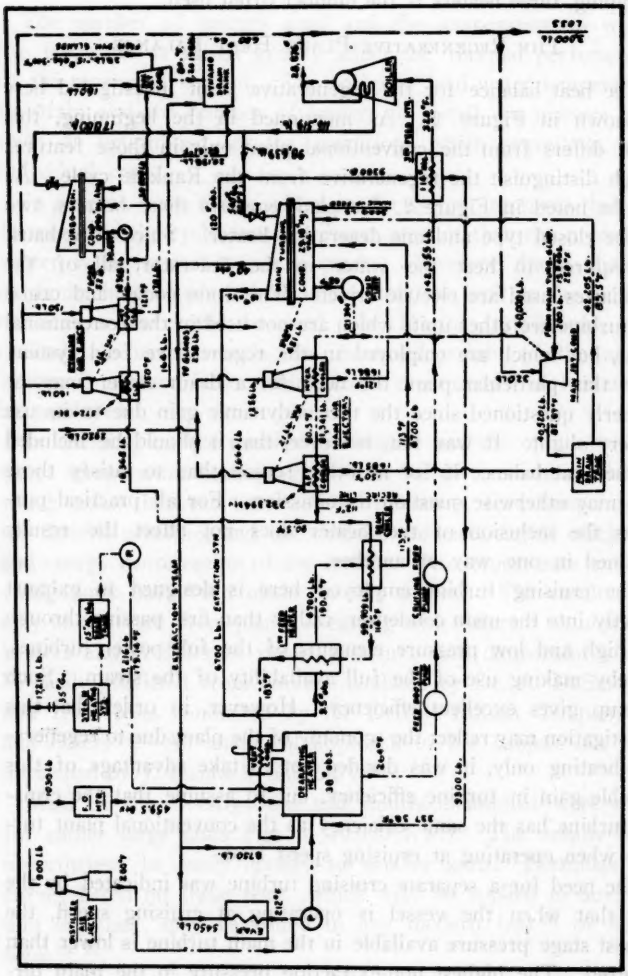
The Regenerative Plant Heat Balance.

The heat balance for the regenerative plant investigated here is shown in Figure 2. As mentioned in the beginning, this plant differs from the conventional plant only in those features which distinguish the regenerative from the Rankine cycle. As will be noted in Figure 2, this plant employs three heaters, two of the closed type and one deaerating heater. Since no exhaust is required to heat the water in the deaerator, all of the auxiliaries used are electric driven. The drain cooler and cruising turbine are other units which are not used in the conventional plant, but which are employed in the regenerative feed system.

In this particular plant the need for a drain cooler may be properly questioned since the thermodynamic gain due to its use is very slight. It was felt, however, that it should be included in the heat balance if for no other reason than to satisfy those who may otherwise question its omission. For all practical purposes the inclusion of the heater does not affect the results obtained in one way or another.

The cruising turbine employed here is designed to exhaust directly into the main condenser, rather than first passing through the high and low pressure elements of the full power turbines, thereby making use of the full availability of the steam. Such a setup gives excellent efficiency. However, in order that this investigation may reflect the economy of the plant due to regenerative heating only, it was decided not to take advantage of this possible gain in turbine efficiency, but to assume that the cruising turbine has the same efficiency as the conventional plant turbine when operating at cruising speed load.

The need for a separate cruising turbine was indicated by the fact that when the vessel is operating at cruising speed, the highest stage pressure available in the main turbine is lower than required. The highest non-extraction pressure in the main turbine at cruising speed (15 knots) as shown in Table I is 64 psi abs., while the maximum required extraction pressure is 120 psi abs., as shown later on.



REGENERATIVE PLANT HEAT BALANCE
Fig. 2.

TABLE I.

NON-EXTRACTION PRESSURES FOR MAIN TURBINE OF CONVENTIONAL PLANT.

Speed of Vessel *Knots*	Pressure——lbs. abs. *1st Stage*	*5th Stage*	*8th Stage*	*12th Stage*
25	390	206	101	32
20	142	77	38	12
15	64	33	16	5
10	36	12	6	1
8	18	5	2	....
5	10	....	....	....

TABLE II.

SALIENT QUANTITIES.

Item	*Conventional Plant*	*Regenerative Plant*	*Per Cent Increase or Decrease*
Fuel to Boilers, lbs./hr.....	10,902	9,768	—10.40
Overall Fuel Rate, lb./shp.....	1.024	0.927	— 9.47
Feed Water to Boilers, lb./hr.	147,609	142,395	— 3.50
Steam to Propulsion Units, lb./hr.	83,547	98,639	+18.00
Steam to Main Condensers, lb./hr.	83,015	68,201	—17.90
Steam to Dynamo Condensers, lb./hr.	15,500	17,880	+15.00
Steam to Main and Dynamo Condensers, lb./hr.	98,515	86,081	—12.66
Steam Extracted for Feed Water Heating, lbs./hr.....	532	24,750	.....

REGENERATIVE PLANT HEAT BALANCE Fig. 2

From published data mentioned above [16], it was decided that the optimum feed water temperature for this plant should be about 327 degrees F., for the initial steam conditions available at the throttle and for the condenser vacuum employed here. In addition, the following rules of accepted practice were adhered to in the heat balance computations.

(1) The steam pressure at the heaters is taken as 5 per cent less than the absolute pressure at the extraction point on the turbine to allow for pressure drop in piping and valves.

(2) The temperature of the feed leaving the heaters is taken as from 5 degrees F. to 15 degrees F. lower than the saturation temperature of the steam in the heaters.

(3) The condensed steam leaves the heaters at the same temperature as the saturated steam in the heaters.

The salient quantities of the heat balance for this plant, as well as the conventional plant, are tabulated in Table II.

Speed Range.

A study of the literature on this subject (as mentioned above) will show that the optimum temperature to which the feedwater should be raised is a function of the pressure and temperature of the steam supplied to the throttle and is also dependent on the condenser vacuum, for any given number of feed heating stages. Furthermore, it is accepted practice to take the temperature of the feed leaving the heater as some value close to the saturation temperature of the steam in the heater. This would indicate that for best economy the extraction pressure of the steam supplied to the heaters must be fairly constant for any given plant. As indicated in Table I however, we see that the stage pressures in a turbine vary greatly with the load on the turbine, so that for a vessel operating at various speeds, it would not be economical to use the regenerative cycle except but for one speed or at most two. Of course if we employ more than one turbine, or if we could get a turbine which would give us the proper combination of extraction pressures for different speeds from various stages, we could arrange to have the vessel operate on the regenerative cycle at more than one speed. Then,

however, we would have to employ a complicated network of extraction piping, and any economy gained would be offset by the complexity of the plant, and the increased weight.

Maneuverability.

In maneuvering, the ship operates for very short periods ahead and astern alternately. In going from ahead to astern, the operation involves closing the throttle of the Cruising Turbine and admitting steam to the astern turbine. It is evident, therefore, that this operation cuts off the bleed steam supply from the feed heaters, rendering them ineffective. This loss of operation of the feed heaters with the consequent loss in economy is not of much consequence as far as economy is concerned, since the periods of operation are of short duration. It is important, however, that the deaerating heater should be operating for even these short periods in order that no air be allowed to enter and stay in the feed system. For this purpose it would be possible to provide a connection from the saturated steam line to the deaerator with an automatic valve which would open and admit steam to the deaerator as soon as the pressure in it dropped down to some predetermined value, as would be the case in going from ahead to astern operation. It can thus be seen that maneuvering while operating on the regenerative cycle need present no more difficulty than is presented by operation on the conventional feed cycle.

Weight Comparison.

It is obvious from comparison of the heat balance computations that for the same shaft horsepower output the regenerative system calls for main condensers, pumps and other units of smaller size than that required for the conventional plant, due to the smaller amount of water flowing through the system, and the smaller quantity of exhaust handled by the condenser. This indicates a reduction in the weight of these items. Furthermore since the regenerative plant uses electric driven auxiliaries instead of steam driven auxiliaries, then a further saving in weight is made due to the absence of the auxiliary steam piping normally

TABLE III.

CONVENTIONAL PLANT WEIGHTS.

(PRINCIPAL UNITS ONLY).

Unit	*Weight (Long Tons)*
Boilers, with accessories	440.80
Main Turbines	128.26
Reduction Gears	223.19
Main Condensers	98.11
Dynamo Condensers	11.30
Forced Draft Blowers	27.50
Main Feed Pumps	15.21
Main Feed Booster Pumps	5.71
Lub. Oil Pumps	3.38
Other Pumps	21.36
Deaerating Feed Tanks	41.40
Turbo Generators	27.10
Main and Dynamo Air Ejectors	5.29
Diesel Generators	30.30
Main Steam Piping	32.70
Auxiliary Steam Piping	44.60
Feed Piping	101.00
	1,257.21

required. These savings in weight are usually great enough to offset the weight of the feed heaters, accompanying piping, and control equipment called for in the regenerative plant. In general then, the regenerative plant is lighter than the conventional plant.

In the case under investigation here however, we have a situation where a direct weight comparison between the two plants is not feasible. The reason for this is that in the conventional

plant all main and auxiliary units may be employed no matter at what speed the vessel is operating, while in the regenerative plant the machinery employed is available for use (in this case) only at cruising speed. Since, as we have noted before, it is not practical to design a regenerative marine plant that will operate economically at several speeds, it is evident that in a naval vessel which has to operate at various speeds, use of the regenerative cycle implies additional propelling and auxiliary units, and therefore, an increase in the total weight of the plant. Table III lists the weights of the principal units employed in the conventional plant. Table IV lists the estimated weights of the additional units required for the regenerative plant when operating at cruising speed. Obviously, the sum of the weights of Tables III and IV represents the total weight of the plant if the vessel under investigation were to employ the regenerative cycle at cruising speed and the conventional feed system for all other speeds. For the plant investigated there is an increase of 9.3 per cent in weight due to the use of the regenerative cycle.

TABLE IV.

REGENERATIVE PLANT WEIGHTS

(ADDITIONAL UNITS ONLY).

Unit	*Weight (Long Tons)*
Cruising Turbines	93.70
F. D. Blowers (Elec. Driven)	3.43
Cruising Feed Pumps	3.39
Cruising Feed Booster Pumps	0.52
Lub. Oil Pumps	1.77
Other Pumps	3.38
Feed Heaters	10.00
Bleed Piping (including Valves)	1.03
	117.22

Note: The weight of the drain cooler has been intentionally left out since, as mentioned in the text, this unit is not considered necessary.

Space Requirements.

It is estimated that each cruising turbine employed in the regenerative plant will require a floor area of 190 square feet, and that each closed feed heater will be approximately 2 feet in diameter, and 8 feet long. It is thus seen that the additional space requirements are considerable, when we consider that the available floor space for machinery in the vessel under consideration is about 3040 square feet per engine room. The space requirement of the cruising turbine alone represents an increase in floor space of 6.25 per cent per engine room.

In fact, the present space requirements in the existing plant are such that it would be almost impossible to find space for the cruising turbine required for regenerative cycle operation. The space requirements of the feed heaters however are not very great. The additional feed heaters called for by the regenerative operation can be easily fitted in the space now available, if vertical heaters are employed. (Due to its restricted nature, the present machinery layout of the vessel under consideration cannot be shown here.)

Summary.

The results of this investigation show that:

(1) The regenerative cycle is more economical than the conventional Rankine cycle. In the vessel under investigation a saving of 9.47 per cent in fuel consumption was affected by use of regenerative feed heating. (See Table II.)

(2) The regenerative plant cannot be employed efficiently over the whole range of speeds a naval warship is called upon to operate.

(3) Use of the regenerative cycle adds to the complexity of the power plant.

(4) For warships, effective use of the regenerative cycle calls for increased weight and space requirements.

Conclusion.

From the foregoing it will be appreciated that from the point of view of plant efficiency and fuel economy, the regenerative

plant is very desirable. In a warship, however, the factors of weight and space requirements, as well as simplicity of plant operation, are more important than economy. Simplicity of operation is so important, in fact, that the Bureau of Ships has only recently decided to eliminate all extraction of steam from turbines because of the complexity introduced, even for warships already in operation for some time.

Therefore, in view of the fact that use of the regenerative cycle on a warship adds to the weight and complexity of the plant, it is questionable whether use of this cycle for naval vessels is feasible.

Acknowledgment.

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The author also wishes to acknowledge the helpful cooperation of his associates and co-workers at the Brooklyn Navy Yard.

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1. "The High-Pressure High-Temperature Turbine-Electric Steamship 'J. W. Van Dyke'" by L. M. Goldsmith, Trans. A.S.M.E. 1939, pp. 205-221.

2. This should not be construed to mean that the conventional plant, where some steam is extracted to heat the feedwater in the deaerator, is also operating on the regenerative cycle. This is not so, since in the conventional plant most of the heating in the deaerator is done by the exhaust from the auxiliaries, and the extracted steam is used only to supplement this exhaust.

3. See "Heat Power Engineering", Part I; Barnard, Ellenwood and Hirshfeld, Chapter XVIII.

4. "A Thermal Study of Available Steam-Power-Plant Heat Cycles" by G. A. Hendrickson and S. T. Vesselowsky, Trans. A.S.M.E. 1934, pp. 203-221.

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11. "The Steam-Turbine Regenerative Cycle—An Analytical Approach" by J. K. Salisbury, Trans. A.S.M.E. April 1942, pp. 231-245.

12. "Heat Power Engineering", Part I, Chap. XVIII, and Part II.

13. "Kent's Mechanical Engineers' Handbook", Part II, pp. 8-76 to 8-80 incl.

14. Westinghouse Elec. and Mfg. Co. booklet "Heat Balance for Modern Power Plants".

15. "The Watts Bar Steam Power Station of the T.V.A." by G. R. Rich and R. T. Mathews, Mech. Engineering, November 1941, pp. 773-779.

16. See Figure 23 of Westinghouse booklet "Heat Balance for Modern Power Plants".

THE EFFECT OF NATIONAL POLICY UPON NAVAL ARCHITECTURE AND ENGINEERING.

By ALEXANDER KIRALFY.

At a time when the great democracies are struggling for survival against the brute forces of militarism it would seem that no pains should be spared to make certain that the basic conceptions underlying all elements of our armed strength be made as scientific as possible.

Offhand it would appear that there is nothing more scientific than a modern warship, embodying, as it does, the highest refinements in the arts and sciences of naval achitecture and engineering and of many associated skills. In the United States and Great Britain the naval sciences have flourished for many years. Century after century the Royal Navy succeeded in defeating all seaborne threats. Not only was the younger American democracy able to add a great contribution to naval history, but, in attaining the position of a leading sea power, it has shown the ability to construct the world's greatest and most powerful dreadnaughts as well as to find a place "second to none" in other ship types.

A little reflection, however, will adduce the fact that there is a substantial difference between the naval sciences and other sciences. The latter aim at the attainment of an unlimited objective, subject only to the availability of knowledge and material, and circumscribed solely by what is feasible. Thus, in a surprisingly few years the airplane grew from a small, hazardous experiment to a transcontinental and transoceanic means of conveyance, its size limited only by the amount of work not merely desirable, but available. The radio grew from a short-range novelty to a global means of transmission. Military art and science reached the point where whole nations became encampments and munition plants, where one natural or man-made obstacle after another lost its hindering influence.

Such appreciated effectiveness, however, has not characterized the naval weapon. It has, in fact, been marked by a decided deterioration. Were a chart drawn covering all arts and sciences, the graphs representing civil and land-power achievements would be found on the positive side of the base-line. But measured in their relationship to the field of war as a whole, the naval graphs would have to be drawn on the negative side of the line. In 1854-55 British and French men-of-war entered the Baltic without compunction and shelled shore establishments. In 1914-18 that had become impossible. In 1914 American and British fleets could have braved Japan in Far Eastern waters. Centuries ago British armies landed easily on a hostile French coast. These have become most difficult enterprises, and quite unthinkable without an overwhelming superiority in the air. The latter involves its own peculiar problems and possible limitations which only experience may disclose, not only in the present struggle but in the unpredictable future. Whether or not the democracies can at all times secure and retain such an aerial ascendancy is a matter of speculation. The " narrow seas " closed to naval enterprise, once almost non-existant, and then limited to the immediate vicinity of coastal fortresses and armed concentrations, have now become synonymous with immense stretches of coastline. And there is no *normal* naval reason in sight why this condition should not be perpetuated. In fact, the whole theory of the Axis geopoliticians has been premised upon the greatly reduced effectiveness of democratic sea power.

This increasing ineffectiveness of democratic sea power may be readily explained. Land-warfare weapons are not manufactured to destroy their " opposites in line ". They are devised to destroy the entire war power of the enemy, including these " opposites in line ", or despite the absence of such matched interference. The German *Luftwaffe* was not proportioned to the French air arm, nor did it content itself with driving the opposing *escadres* out of the air—it struck at every desirable objective. Military objectives on land are unlimited.

At sea, however, the naval objective has been seriously limited. All developments to the contrary notwithstanding, democratic navies, in modern times, have been proportioned to those of the land powers—to those of France, and subsequently to those of

Germany. As the land powers were primarily concerned with their armies, it is patent that they did not devote much effort to their navies. To an appreciable extent this observation applies to Japan as well. Consequently the measure of the naval strength of the democracies—their *prime* concern, has been the *secondary* efforts of the land powers. Democratic navies were therefore built solely to meet their "opposities in line", and not to achieve total victory. We came to believe that the latter goal could not be attained by means of sea power.

However, a survey of history disclosed the fact that this limitation has not been inherent in the naval arm, and hence inescapable. It has flowed from the democratic non-aggressive concept of sea power as distinguished from the militaristic aggressive concept of sea power as evidenced by the decisive naval activities of land powers in bygone years and, more recently, by the pronounced lead they have taken in technical naval development. This dual conception of sea power has been obscured by the fact that, whereas there is voluminous evidence of the comparatively recent activities of democratic sea power, decisive displays of autocratic sea power have been few. Yet on these few occasions militaristic land powers were able to destroy democratic sea powers. Such was the fate of Phoenicia, Carthage and Greece. Medieval Europe was thrown into chaos by sea-borne Vandals, Moors and Vikings. The "river power" of the Mongols conquered China. In modern Europe the British Navy found itself facing increasing difficulties which nearly culminated in disaster in 1917. The combined sea power of the United States and Great Britain today face a grave dilemma.

In an article in the June, 1941, issue of the *United States Naval Institute Proceedings* the writer attributed this condition to the fact that autocratic navies had made the enemy's homeland their basic target, but that they did not generally construct warships intended to be used against such targets until after one land power had subjugated all neighboring potential land enemies. During the respite thus experienced democratic sea power was able to flourish for the reason that it had not been subjected to the all-out assault of autocratic sea power. It had merely been called upon to meet the more normal types of warships which the land power enemy, pending the fruition of his schemes on that element, had deemed it necessary to build.

Militaristic sea power appears to have followed two distinct, consecutive stages of naval warfare. During the preliminary stage, when the aggressor was concentrating his attention upon land ventures, he usually developed unorthodox methods of containing or counteracting such democratic sea power as might interfere with his immediate plans. These methods, in recent years, involved the employment of the mine, torpedo, air bomb and improvements in coast and mobile batteries. Once an hegomony was established on land, the aggressor has shown an ability to construct warships of remarkable qualities intended to be used, first against "normal" squadrons of the democracies, and then against their coasts. The *Bismark* fell into the one category, the Japanese Navy into the other, though not to a pronounced degree inasmuch as its targets in the Far East were militarily weak. None the less its emphasis upon aircraft carriers, barge-transports and armored landing boats, etc., is significant.

Had the militarists been naval powers constrained to reach their enemies by sea, it is patent they would have confined themselves to the last stage of naval war. But inasmuch as the democracies *are* in that position, it would appear that the time has gone by when they can any longer overlook the sound implications of this land-power naval outlook, an outlook that destroyed too many democracies, ancient and modern, to warrant our taking the risk of its being redeveloped by militarists. And it follows as a corollary that the democracies might well consider the use of "second-stage" warships against their present and future enemies.

In such an endeavor the full burden would naturally fall upon the shoulders of the naval architect and engineer. The challenge appears to be as great as it is unparalleled in history. To convert fundamentally defensive ships into the offensive type the shortcomings, not of decades, but actually of many centuries, would have to be recouped.

In democracies the architect and engineer have rarely been called upon or even permitted to devise aggressive "anti-land" ships. National policy, and not professional limitations, compelled them to build the kind of fighting ships to which we have grown accustomed. Revolutionary changes in naval construction and equipment should, it is believed, be based upon an appreciation of the handicaps under which professional naval constructors in the

free countries have been laboring since time immemorial. The greatness and continuity of this handicap will be the measure of the ground that would have to be recovered. Whether and when such revolutionary changes can be put into effect are matters of industrial and political expediency, not of scientific soundness. There can be little question, however, but that this soundness should be realized, whatever action might be taken as a result of such realization.

The moment land is looked upon as the prime target and hostile navies as secondary targets, as impediments in the way of reaching the more "military" objectives, an unlimited conception of sea power will obtain. Thoughts will flow away from the standardized types of capital ships, cruisers, aircraft carriers and so forth. The principal unit of the new sea power would probably be an immense coast-attack ship whose efforts would be seconded by similarly large aircraft carriers, engineering ships, equipment vessels, including tank-carriers, and troopships. All would have to be so large and so heavily armored as to withstand every form of naval attack and to be immune to normal air attack. There are no technical limitations to increases in size (1), and the matter of armor protection against torpedoes, for instance, has been discussed long ago. (2). In such a scheme the submarine and destroyer might disappear as types and the utility of the cruiser be considerably diminished.

The above types of "anti-land power" ships are mentioned purely by way of suggestion. It is the prime purpose of this discussion to show the hold that defensive considerations have had upon democratic naval construction and the manner in which the land powers, not so "inhibited", have led the way in the development and employment of aggressive ship types. To repeat, it would seem that a survey of these handicaps cannot help but improve the technical atmosphere, so to say, and place the technician in the democracies in the unfettered position long enjoyed by those in the autocracies.

In reexamining history from a technical standpoint it will be discovered that democracies tended to be extremely conservative in their naval construction and to rely, for defense, upon theories which stemmed from a consideration of the individual seaman rather than from a consideration of the inherent capabilities of

the ship as the decisive unit. They have depended more upon skillful " blue water " maneuvering than upon shock tactics. They insisted upon pitting the navigator against the soldier. The ship thus became the servant of the seaman and there was developed a strong democratic tendency to resist any innovations that would either render the ship so " scientific " that the human element might become confused, or convert it into what would be deemed an instrument of aggression.

Among militaristic powers the ship was master and the seaman servant. Victory was to be secured by the instrument rather than by the hand that manipulated it. This placed emphasis upon ruggedness of construction and upon the quick acceptance of inventions which would increase the effectiveness of that ruggedness. For the individual aptitudes of free seamen was substituted a continuous appeal to the refinements of science. It must be admitted that, in the long run, and despite intermittent set-backs, scientific soundness must prevail. Let us therefore consider the effect of these opposing naval ideologies through the ages, and the back-log of anti-scientific resistances built up in the democracies.

Although the Greek expedition against Troy is believed to have taken place in the thirteenth century B.C., the open boats then used showed little improvement some nine hundred years later. It has been observed that these progressive democrats " up to the fourth century B.C. appear to have contributed nothing to shipbuilding." (3). For long there was no " defensive " reason why the Athenians—the Greek naval element, should have improved their standard warships, which may have differed but slightly from their merchantmen. By the sixth century B.C. the democratic Phoenicians had already taken a definite step forward by arming their warships with the beak or ram. This is brought out by the heavy, deep-draft beak shown in Figure 10 of Torr's work (4) and may have been borrowed from the ancient Egyptians. (5).

At first the Greeks seem to have adopted a long, slender ram that was a protuberance rather than part of the ship itself. It is described by Torr (6) and illustrated in the handbook of Jules Vars. (7). This type of ram is peculiarly " democratic ". It involved great skill in navigation coupled with a " minimum risk " to the ship to which it was affixed, a distinctly defensive concept. Though, by the year 600 B.C., the Greek ram had grown sturdier

(8), it remained a more or less separate fixture, and more than a century later is shown to be still quite allongated. (9). A glimpse of the same line of reasoning may be caught in the manner whereby, in recent years, we removed underwater torpedo tubes from our capital ships, the offensive arm being sacrificed to secure greater defensibility.

Militarists often profit more from their defeats than democracies do from their victories. At the Battle of the Corinthian Gulf in 429 B.C. the Greek land-powers, which may be referred to compositely as Sparta, tried to take advantage of the strength of their stems and rams by "forming circle" outwards against the Athenian attack. The dexterity of the Athenian seamen, however, won the day. Thereupon the Spartans strengthened their stems and rams, incidentally lowering the position of the ram, and so defeated a small Athenian squadron. Shortly thereafter Athens was brought to her knees at Syracuse by a defeat incurred at the hands of the "strong-stemmed" militarists. By their improvement the Spartans had actually increased their "armor" thickness and their "gun-power". Science and shock tactics had overcome individual skill and prowess. (10).

Rome, essentially a land power, was the only democracy that appears to have understood aggressive naval warfare. It will be recalled that the Romans destroyed the thousand-year-old sea power of Carthage by means of the *corvus*, a gang-plank contrivance whereby the skillful Carthaginian navigators were prevented from ramming and compelled to enagage in land-warfare tactics at sea. (11). The Romans then built sturdier ships equipped with rams of greater power which were generally part and parcel of the ship. (12). Mention need merely be made of the "coast attack" ships by means of which Alexander of Macedon conquered Tyre and doomed Phoenicia. (12A). In all cases the superior offensive architectural and engineering developments of the land powers eradicated the sea power and hence the life of the three great republican naval powers of the ancient world.

The decking of warships, giving them added ruggedness, protection for the "engineering department" of rowers and a loftier platform for the "artillery" of archers, seems to have begun with the land powers, such as Egypt, to have been passed along to the Phoenicians, and then to the Greeks. While it may not be

entirely correct to state that none of the Greek ships at Salamis (413 B.C.) was fully decked (13) they were substantially inferior in military qualities to the Persians (14), showing the influence that land power had on the shipbuilding of the Phoenicians and others in its service. There is little evidence that the Greeks adopted turrets and projectile-throwing engines, though the Romans made good use of these contrivances. (15). The equivalent of side armor, in the form of raised bulwarks, existed in Egyptian ships. (16).

Mention should be made of the eagerness with which land powers adopted "armored" loop-holed cabins. They were employed by Cyrus of ancient Persia in his Caucasian expedition (17) and by Belisarius in his attack on Rome. (18). They were contrived in Central Asia in 1218 (19) and reappeared with much significance in the French floating batteries which failed before Gibraltar largely because of remediable defects. (20).

As the Dark Ages rolled back, republican Venice emerged as the great sea carrier and placed reliance on a galley in which fragility of construction was coupled with the customary democratic skill at maneuvering. By about 1275, however, the ships of the land power Spain rivalled those of the Italians in *stoutness* (a word far more frequently found in land power writings than in those of the sea powers). (21). In a brief space of time Spanish galleys are referred to as *colossi* and as "lords of the Mediterranean". (22). In this period the Portuguese launched the *S. João Baptista* of 364 guns, believed to be the world's greatest galley. (22).

British sea power was long characterized by the unwillingness of the Briton to construct and maintain a war fleet, and this attitude resulted directly in the Norman conquest. Until the fourteenth century Britain's navy consisted almost entirely of armed merchantmen. There was the urge to keep costs down, to shun improvements that would inure more to the benefit of an aggressor than to the non-aggressive Briton. The French and Spanish land powers led the way in the building of powerful ships. At the Battle off La Rochelle in 1372 "The King of Castile" had at his disposal ships of a size and strength unknown in other countries, formidable opponents to the petty trading vessels which constituted the greater part of the English Navy. (24). These

ships mounted guns and probably other engines of destruction, and inflicted a severe defeat on a British force. France followed suit by building ships of what was then deemed to be an "enormous size". A fleet of these vessels successfully held off a numerically superior British formation until the arrival of a Spanish squadron brought success in its train.

From this period until the reign of Elizabeth the British repeatedly copied French ships, France's *Marie la Cordelière* having been in 1512 the largest warship afloat. It was now that Sir Walter Raleigh complained that his country's men-of-war were lacking in strength of construction, stoutness of scantlings, speed and the ability to fire their guns in all weather. (25). The port-hole, which permitted the incorporation of more than one gun-deck, was invented by Descharges of Brest. (26). As late as the end of the 18th century the French, and not the British, could fire all their guns in all weather, thanks to the raising of the lower battery. (27).

Particular significance attaches to the report of Le Sieur Arnoul dispatched to Holland and England in 1670 by the French navy minister Colbert. (28). He advised his superior that the ships of democratic Holland were not constructed as strongly as were those of France, that, because of the treatment of the wood, shots which would not pierce a French warship would perforate the woodwork of a Dutchman. He pointed out that, being commerce-minded, this republic used its warships for trade, and hence designed them to carry appreciable cargoes. The Netherlanders, on the other hand, criticized the weight and poor sailing qualities of the French. Arnoul stated that the timbers in British ships were not quite as heavy as the French, and that the Royal Navy used far less ironwork for knees, etc. The considerable fore and aft overhangs of English ships, though they facilitated good sailing, were held to be militarily objectionable.

It was another Frenchman, Paul Hoste who, in 1696, first applied the theory of scientific mechanics to the construction and management of vessels. In an extremely interesting work approved by the Académie des Sciences in 1764, M. Bourde Villeheut (29) complained that the recently built French ships were not as strong as formerly, and so could not so well withstand enemy attack—a relationship between "armoring" and enemy attack

rarely found outside of land-power thinking in the days of sail. He wrote that " In older ships at most places shot could not go through ". He argued that gunners should be better protected against " superior forces ", and that masts should be strengthened in this regard. His work is outstanding in that a whole chapter is devoted to the matter of " stoutness ", a matter almost totally disregarded in Charnock's three volumes, who seems only to have been interested in " solidity " when showing how French ships, despite that feature, were handier than British. (30).

The bomb-ketch, the fore-runner of the shell-gun vessel, and hence of the modern capital ship, was introduced by the French—Bernard Renan, about 1679 (31), when a 200 pound bomb was injected into a world where the 32-pounder was the norm.

Similar conditions obtained during the eighteenth century when " It may be truly said that . . the majority of the improvements introduced in the forms and proportions of vessels of the Royal Navy were copied from French prizes." (32). In 1706, according to the same source, every French type was superior to corresponding British types, in this case as to sailing qualities and the number of guns carried. To meet the Dutch criticism of their speed the French contrived to increase this factor by science, not by the sacrifice of stoutness. In 1741, when the genuine frigate was introduced, the Frenchman was the larger, and British dimensions had again to be increased in 1788 to " keep pace with the improved French and Spanish ships." (33).

The introduction of steam encountered strong democratic misgivings, despite its acceptance commercially, so that the French " were already building long and narrow steamers of superior speeds under steam, but with no sail power, while in Great Britain we were building short, protected, well-armed ships, with considerable sail power, but not as fast as steam ships." (34). France embraced steam, " hoping to find in an 'industrial navy' compensation for her lack of seamen." (35). This brings out not only the age-old desire of the land powers to act by ship rather than by the individual, but to overcome any superiority the democracies had by virtue of their sea-going populations. In fact one British officer, to mention no others, complained that " during the last century we had more skillful seamen than other nations possessed " and that " the handling of a steamer can never need the practiced

eye and skillful, almost sympathetic direction of the true sailor, who has been trained in ships under canvas only." (36). Lord Melville felt that steam would strike a fatal blow at the naval supremacy of the empire. (37).

Steam naturally deprived the democratic seamen of their "weather gauge" and of the ability to cruise for months without "refueling". While care was taken to economize coal, to preserve cruising radius (v. Admiralty circular No. 177) and there was a strong urge to preserve the sailing qualities of warships, there remained, as ever, the desire to cultivate "great vigilance, sound judgment, prompt decision, steady nerves, physical agility, grit and daring" on the part of officers and men. (38). And there was the fear that something might go wrong with the engines. Consequently, though in 1822 M. Paixhans advocated a French fleet of steamships armed with shell guns, wherein "steam was to be the master", Sir John Ross, in 1828, desired that steamers be assigned to sailing warships to tow them into positions of vantage, steam being made the servant. (39). So great was the resistance to the use of steam that attempts were made to propel a vessel mechanically by man power. (40).

The influence of steam upon naval architecture was also pronounced, serving but to increase democratic hesitations. Paddle wheels were vulnerable, besides which they interfered with the sails and with broadside armaments. One need but speculate as to the fate of a Nelson trying to "break the line" by means of paddle-wheelers at a steamship Trafalgar. Further resistance met the introduction of the screw propeller in its various forms. During experiments in 1856-7 the four propeller blades of the *Bullfinch* were wrenched off the boss simultaneously, the vessel acting as though affected by a submarine earthquake. (41). Obviously, in a democracy whose very life depended upon successful naval operations, the vision of a fleet losing its propeller blades at the height of action was something that could not be contemplated without grave misgivings. The seaman could always be relied upon to fight for his country, but mechanical contrivances seemed to be not so "sympathetic".

An increase in the number of screws was made necessary by the desirability of "keeping cylinder heads below the protective deck". It seems that the French were concerned with this "mili-

tary" feature whereas in the United States and Great Britain attention was concentrated on speed and engineering efficiency. (42).

Similar observations apply to the water-tube boilers, patents for which were taken out in France by the Générateurs Belleville in 1850. (43). The experience of the *Bullfinch* was repeated when a Herreshoff boiler was mishandled and exploded on shipboard at Portsmouth in 1879 because it had been installed "without reckoning upon the possible attitude of the working engineers and stokers in the Navy to a system which enormously increased their cares and responsibilities." (44). It was also feared "that the high pressure boilers of the triple-expansion type would give much more trouble, both in the manufacture and in working at sea, than those of the older type." (45).

Conflicting ideologies also manifested themselves in the matter of forced draft. In the Royal Navy it appears that doubts as to the efficacy of the mechanism held back its employment, with the resultant large openings in the armored deck for the air shafts. The French, on the other hand, thinking more in terms of the integrity of horizontal protection, utilized forced draft. (45A).

If misgivings arrested the development of the steam engine on shipboard, the introduction first of armor and then of iron ships were met with veritable dismay. In fact, the first British reaction to shellfire was to seek means of rendering wood fireproof rather than shielding it with iron. (46). It should be noted, however, that London appears to have taken the lead in *designing* iron ships. (47).

The Admiralty opposed the development of iron in naval construction because of the fear that it would tear great gaps instead of clean holes that could be readily plugged, and one of the main reasons why the new material was adopted was because wooden sterns could not stand the action of the screw shaft. (48). But the underlying reason for this resistance was that wooden ships were rarely sunk in battle whereas it appeared inevitable that badly damaged iron vessels must go to the bottom. Another objection in Britain was that iron would reduce velocity and seagoing qualities. (49). In 1850 the conclusion was derived from experiments that metal was not suitable. Consequently, when evaluating the efforts of M. Dupuy de Lôme, the earliest champion

of iron over wood in France, the British Director of Naval Construction stated that "in boldness of conception . . . he takes the first place among the naval constructors of our time." (50).

The advent of steel again found the land power, France, ahead of the naval democracy, Britain. Said one authority, "To the French belongs the credit of having been the first to recognize the advantages of mild steel for shipbuilding purposes, they having introduced it into portions of the structure of warships in the early part of the year 1873. (51). Thereafter the Director of Naval Construction of the Admiralty challenged the steel makers of the United Kingdom to produce a material equal to that manufactured in France. (52). Sir Nathaniel Barnaby himself wrote on the subject of French naval construction to the effect that in October, 1874, he was bound to say that "there was an extended use of steel which argued greater confidence in the material than we had ourselves." (53).

The application of armor to the sides of ships had already produced the navigator versus soldier controversies. It was feared that armored sea-going ships would behave badly. (54). As is well known, the first armored ships, other than the so-called freak experiments of early days, were the floating batteries which Napoleon III employed in the Crimean War. He next led the way with the *Gloire* wherein, according to a British authority, "the instincts of the engineer, familiar with land fortifications, prevailed over the seaman." (55), a sentence intended to embody a criticism, but unconsciously striking at the roots of the distinctions between offensive militaristic naval power and the defensive conceptions of the democracies. This is well brought out by the same writer who stated that armor was "contrary to all the seaman's instincts and especially to the instinct of the French seaman." (56). He attempted to support this contention by pointing out that the French were then firing their heavy guns over barbettes rather than out of turrets, so that the gunners "could see what they were doing." He was obviously trying to identify such action on the part of the French seamen with "the unreadiness of the British sailor to give attention to anything except what is hurtful to the enemy" and the unwillingness of this seaman to "adopt or to improvise anything which seems directed towards securing his own safety." (57). The democratic writer, instilled with his con-

cept of the ascendancy of the man over the material, and forgetting that safety measures would be equally applicable to the ship itself, failed to glimpse the truth of the matter. The French fired their guns *en barbette* for a very "military" reason. Their designers "did not like enclosed turrets because these had to be brought back into original position for reloading". (58). Contrary to the English interpretation of the matter, the French were willing to sacrifice men to secure more rapid firing, i.e. to make the ship a more efficient fighting unit. When, during the last years of the broadside era, the British took steps to increase rapidity of fire, they reduced the size of gun from a 42 to a 32-pounder, rather than devise means of securing the benefits of the heavier shot. (59).

While, in modern times, there has been neither the opportunity nor the occasion for land powers to embark upon all-out assaults upon the naval democracies, the concept of the land as the prime target, with the consequent necessity of a "stout" ship, none the less continued to pervade land-power thought. Worthy of quotation are the words of Hauser himself—"For offensive action the fleet will bombard fortified positions and cover the landing of an army. For this purpose it will require strongly protected ships provided with powerful artillery, and able to penetrate shallow waters, as well as transports." (60). According to Vice Admiral Jurien de la Gravière, "The navy of the future, in my opinion, will be one able to open up to the strongest armies a highway to the enemy's capital." (61). Even in a pre-World War I German handbook space was found to indicate the co-operation of a fleet with the army in landing on hostile shores, whereupon the fleet would become the principal base *(Hauptbasis)* for the expedition. (62). In 1929 one of Germany's leading naval authorities wrote that "One of the most important means of exercising sea power is in combined operations." (63).

In opposition to this "soldier's" outlook it would be difficult to find a plainer exposition of the "seaman's" view than the lines penned by Sir Charles Elliot in 1865; which seem to merit quoting at length:

"You may build a ship as big as St. Paul's, as invulnerable to shot as the Rock of Gibraltar, and with an armament capable of sinking Beachy Head . . . (but) you must convince me, first, that

she can contend with the state of the sea in its anger round Cape Horn and the Cape of Good Hope, secondly, that she is as handy and as fast as the Queen's yacht, and thirdly, that she can float on the waters where our occasions call us." (64).

Except for the latter point, noted by Hauser, and obviously of great importance, the reasons given are navigational rather than military, and the hazards of the elements can usually be avoided by building larger units, if not by actually avoiding tempestuous regions temporarily, though these are few and not found in the immediate vicinity of Europe in particular.

Some mention has been made of the matter of ordnance. The introduction of the horizontally firing shell-gun by the French and its historic use in the Black Sea by the Russians compelled Britain to follow suit. Resistance to the innovation came in the form of an unwillingness for many years to employ the "complicated" breech-loader—it might jam in battle. Even after its introduction the muzzle-loader was reverted to, only to be dropped again, principally because of the advent of a new land power into the domain of naval warfare—Germany (65), together with the development of slow-burning explosives. (66). Whereas, in 1865, the *Gloire* and other French ships were armed with rifled breech-loading guns (67), the British ships of 1870 "had only short muzzle-loading guns, firing a projectile with a low velocity and mounted so as to fire out of a small port, giving a very limited angle of training. Until (1888) we were very far behind the breech-loading guns of both the French and Krupp." (68). Even the rifled gun was looked at askance, the clogging of the grooves having occasioned concern. (69). In 1885 when London feared trouble with St. Petersburg all of Britain's thirteen best battleships carried muzzle-loaders whereas the big ships of the Russian land power had breech-loaders. In 1867 both the French and German armor-clads were fitted with breech-loaders. (70)

The rapprochement of France and Britain resulted in the French concentration on land with the resultant deterioration of interest in the sea from an offensive standpoint. Other land powers, notably Russia, devoted their attention to such circumventing devices as the mine and torpedo, later followed by Germany. The French tendency towards "navigator's" ships became pronounced, as seen in later years by the development of extremely

speedy cruisers and destroyers and the building of particularly large and fast flotilla leaders. The ability and desire to attack a land-based enemy rapidly dwindled. In 1906 one Frenchman bitterly complained that "the strongest armored ships must flee a defended coast as it would the pest, lest it find a grave in its waters." (71). And thinking, as Briton's rarely thought, of a hostile shore, Admiral Sir R. Hastings admitted in 1913 that "While our old-time wooden battleships could with impunity cruise day after day within sight of a hostile port, our misnamed 'Dreadnaught' is compelled even by day, to give it a wide berth . . ." (72)

It might be fitting at this juncture to advert to the ruggedness of the Kaiser's High Seas Fleet, to the fact that no German ships blew up by reason of magazine explosion, and to the report that it took six torpedoes to scuttle a Russian battleship to prevent it falling into German hands in the last war. (73).

When we come to the rise of American sea power we find a remarkable spectacle wherein the age-old British inhibitions against aggressive ships are found duplicated, but coupled with an admirable willingness to try out new theories and devices. This may be explainable by the fact that, while the United States was not close to powerful enemies nor susceptible of being jeopardized if not actually destroyed by a naval defeat, the pioneering spirit continued to hold sway. London always recognized the need for a navy, but was chary about altering it. In America the navy was at times sadly neglected, but when one was constructed, it was infused with the vision that has made this country what it is.

In writing of the frigates authorized by the law of March 27, 1794, Cooper pointed out that "An important and recent improvement in ship building was adopted, by which frigates were increased in size and efficiency". (74). Referring to the following decade he informs us that "Ships that rated eighteen, instead of carrying sixes, or nines, or even twelves, began to carry thirty-two pound carronades, and they required greater strength, thicker bulwarks, and larger ports than it had been the custom formerly to give to vessels of this class." (75)

In describing the U.S.S. *Constitution*, Lieut. Commander Frost not only shows her great "stoutness", but records an English objection thereto which is strangely reminiscent of the Dutch

criticism of French ships in the days of Colbert. "We had gone to the 24-pounder gun while the British retained the 18-pounder. This gave us an advantage in weight of metal thrown by one broadside of 100 to 70. However, the British still believed that the 18-pounder was the better gun for frigates and that we had weighted down our ships with too heavy a battery. Again, our hulls (i.e. for frigates) were built very strongly—were one inch thicker than those of Britain's 74's (ships of the line). Here again our British friends believed that they would be dull sailors. They showed not the slightest concern over these American advantages." (76). This officer went on to say that "Yankee gunners are the best in the world" (the emphasis upon the individual), and that "they had iron sides to protect them: twenty inches of stout oak beams"—the land-power concept of the ship. Mahan has observed how the *Constitution* holed the *Guerrière* below the water-line. (77)

Informative in this regard is a letter dated July 2, 1798, from the Secretary of the Navy to the effect that "Frigates carrying 38 guns in the British Service, the largest sized Frigates they employ, measure not exceeding 940 Tons. The (American) Frigates heretofore built-the *Constellation* to carry 36 Guns, but actually carrying 38, measures upwards of 1300 Tons". (78). Several days later this official wrote of "the essential Qualities of Swift Sailing and considerable *Strength*". He went so far as to conclude later that "The Frigates heretofore built have been on too large a scale. . ."

The introduction of steam, iron and armor into the United States Navy are too well known to require elaboration. It will be recalled that Fulton's *Demologos,* launched in 1814, was not only the world's first war steamer, but that it embraced a number of extremely novel features, incuding four-foot sides. When the screw propeller was adopted in Britain in 1845, it was with the condition that it could be unshipped. (79). The first screw ship of the line was not launched there until 1852. (80). In the U.S.S. *Princeton,* launched in 1843, the propeller shaft was for the first time placed below the water-line. (81). This war steamer was followed by six other screw frigates, including the *Roanake,* launched in 1863, which was a turreted frigate. (82). Mention need merely be made of the epoch-making characteristics of the

Monitor and the fact that, although it foundered at sea in December, 1862, more of this type were built, culminating in the seagoing *Miantonomoh* class. (83).

British experts have not hesitated to acclaim American naval progress. The following notes are from the already much-quoted works of Sir Nathaniel Barnaby and Sir George C. V. Holmes:

"Revolving armored turrets, originated in the United States, and developed (in England) by Captain Cowper Cowles." (84)

"Improvements in the dimensions and forms of our ships were only carried out after they had been originally adopted by the French, or Spaniards, or more recently by the people of the United States". (85)

"In the manufacture of armor the United States factories have forced the pace in inventive progress". (86)

The same authority remarked that although United States warships in 1892-1900 were fitted with cofferdams filled with obturating material and that France had experimented similarly with cocoa-nut fibre, the British Admiralty was not known to have done so. (87). This feature has been commented upon by Professor Hovgaard. (88). Again it is merely necessary to make mention of Dahlgren's contribution to the manufacture of ordnance (89) and to Admiral Fiske's achievements in the field of gunnery. (90).

As previously indicated there were the usual determents in this country. In 1865 Isherwood, Chief Engineer of the United States Navy, complained of the difficulties he encountered in securing acceptance of his theories of superheated steam. (91). Despite the success of the *Princeton* we continued to build paddlewheelers. (92). The contract with Ericsson for the *Monitor* stipulated that he would have to furnish "masts, spars, sails and rigging"—though this condition was neither adhered to by the inventor nor inisisted upon by the Navy Department. (93). We have questioned the practicability of the diesel engine for large surface ships by limiting consideration thereon to small units (94), though they were installed in the German pocket battleships, a country that could, while devoting its full attention to land enterprises, afford to take greater liberties with its fleet, and similar observations might apply to increased steam pressure. (95). In 1935 a Naval Academy textbook referred to "those

who have long urged damage control on seemingly unhearing ears"—the "unhearing ears" being suggestive of the British seaman's unwillingness to ponder his own safety. Two score years previously some doubts had been entertained regarding the principal feature of damage control, i.e. watertight compartmentization. (96). The manner in which the Germans led the way in welding is, of course, well known. Our realization of the anti-land weakness of warship construction was touched upon recently when a naval officer uttered the truism that only "a very dire necessity would induce any naval commander to take his fleet under the fire of modern shore batteries". (97). He was undoubtedly referring to permanent shore installations which "coast attack" ships could avoid, though in the final analysis these would be one of its logical targets.

All factors considered, the United States has been outstanding by the manner in which it trended upwards the concept of all-out naval war. It has disregarded speed in its capital ships in favor of armor, of which its ships appear to have the best protection of any in the world. The U.S.S. *Iowa* is, insofar as can be known, the largest and "stoutest" warship ever built. There can be little question, therefore, but that a reexamination of the significance of sea power and its weapons could be undertaken in this country with every hope that such an investigation would be highly productive.

It is believed that the above will indicate, even if only in a brief sketch, the ageless tendency of land-powers to construct more "warlike" vessels than has been the case with the democracies. Despite his reliance upon the land and aerial arms and the submarine, the Nazi *Führer,* too, has disclosed his interest in "stout" and heavily gunned ships. (98). It is furthermore believed that the absence of "coast attack" ships in modern fleets should not be emphasized inasmuch as, since the Middle Ages, no aggressive land power was able to achieve that domination of Europe which would enable it to go all-out from a naval standpoint. What such a land power would do may be sufficiently glimpsed through the course of history.

The localization of war, the successful cooperation of friendly land-powers and the grip of the blockade can no longer guarantee the safety of democratic naval powers. We have no assurance

but that air power might prove to be a double-edged weapon. Sea power that can be employed against hostile land positions is not only scientific, but possesses the means of bringing strong pressure to bear during peacetime upon would-be aggressors. But whatever the potentialities of unlimited sea power, these cannot obviously be brought out except and until we realize that our naval conceptions have been adversely affected by ideological interferences which, in the final analysis, do not belong in the realm of national security. We would not for a moment think of being ideological about tanks and bombers any more than we would be ideological about railroads and medicine. Yet, as has been shown in the foregoing pages, we have been ideological about the shape and sizes of our warships, about their ordnance and propelling power, and about the stuff whereof they have been fashioned. Once the naval architect and engineer is relieved of these non-technical impediments democratic sea power could most likely acquire a decisive significance which it has hitherto not enjoyed. What concrete forms such a sea power would take is a matter which the writer must consign to more capable hands than his.

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THE HULL AND ITS SCREW PROPELLER.

By E. A. Stevens, Jr., Member.

Parts 1 and 2 of this article were published in the August 1942 issue of the Journal.

Part 3—The Propeller.

Basic Conditions and Methods Used for Determining the Shaft Horsepower and Revolutions Per Minute.

METHOD USED.

The method used in estimating the performance of a vessel when the data of the hull, effective horsepower and characteristics of the propeller are given is:

First, to obtain the shaft horsepower, effective horsepower, revolutions and speed, assuming the propeller is operating under Basic Conditions.

Second, correct the shaft horsepower for the difference between the actual and basic effective horsepower.

Third, correct the revolutions for the actual conditions. In this there are two corrections. First, the correction for the difference between the actual and basic shaft horsepower; and, second, for the difference between the actual and basic speed.

Design problems are the reverse of the above and will be given later in the part dealing with such problems.

DEFINITION OF TERMS AND ABBREVIATIONS.

N = number of propellers.

SHP = basic shaft horsepower on one shaft

SHP_d = actual shaft horsepower on one shaft

R = basic revolutions per minute

R_d = actual revolutions per minute

D = diameter of propeller in feet

d = diameter of the propeller hub in feet

P = pitch of propeller in feet

PA ÷ DA = projected area ratio of 3 blades = total projected area ratio of a 3-bladed propeller, also 3/4 total projected area ratio of a 4-bladed propeller, and 3/2 of the total projected area ratio of a 2-bladed propeller

4/3 PA ÷ DA = total projected area ratio of a 4-bladed propeller

2/3 PA ÷ DA = total projected area ratio of a 2-bladed propeller

ST_d = indicated thrust per square inch of disc area

$$= (33000 \times SHP) \div (P \times R \times \frac{D^2\pi}{4} \times 144) =$$

$$(291.8 \times SHP) \div (P \times R \times D^2)$$

T.S. = tip speed in feet per minute = $R \times \pi \times D$

V. = basic speed in knots

v. = actual speed in knots

E.H.P. = basic effective horsepower on one shaft.

P.C. = basic propulsive coefficient = E.H.P. ÷ S.H.P.

e.h.p. = actual effective horsepower on one shaft

p.c. = actual propulsive coefficient = e.h.p. ÷ $S.H.P._d$

BASIC CONDITIONS.

As Admiral Dyson has explained in his books on screw propellers and in several articles in this JOURNAL, how the basic values of propulsive coefficients, indicated thrusts, tip speeds and the power correction factor have been obtained, these explanations will be omitted.

In order to facilitate calculations, Table I has been prepared which can be used in place of Chart 6. In this table are given values of $S.T._d$, T.S. ÷ π and P.C'. The value T.S. ÷ π has been used in place of T.S. as whenever tip speed is used in any

CHART 6

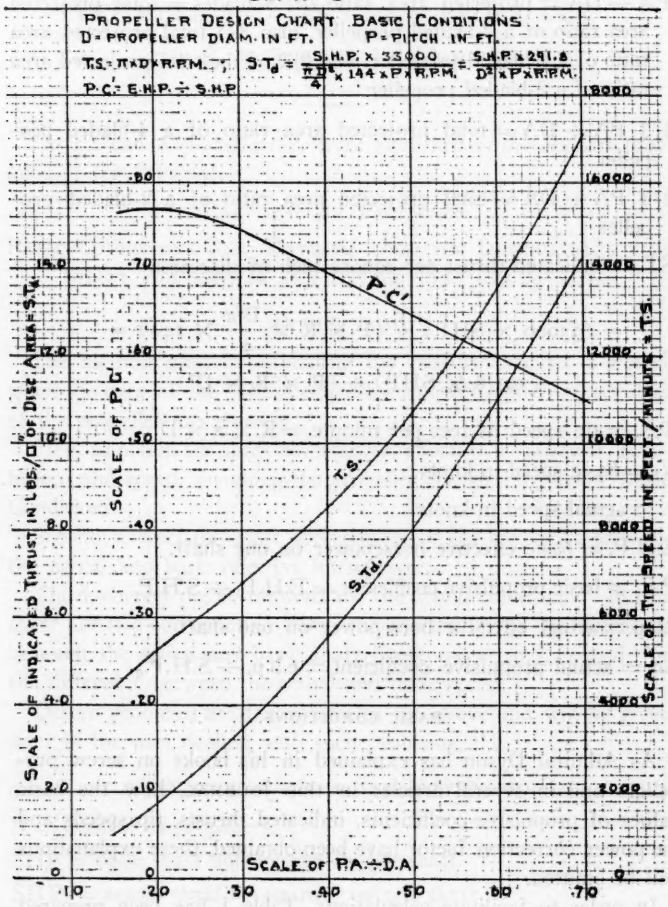

PROPELLER DESIGN CHART BASIC CONDITIONS
D = PROPELLER DIAM. IN FT. P = PITCH IN FT.
T.S. = π×D×R.P.M. ; S.T.d = S.H.P. x 33000 / (π/4 D² x 144 x P x R.P.M.) = S.H.P x 291.8 / D² x P x R.P.M.
P.C. = E.H.P. ÷ S.H.P
P.C.
T.S.
S.T.d.
SCALE OF P.C.
SCALE OF INDICATED THRUST IN LBS./□ OF DISC AREA S.T.d.
SCALE OF TIP SPEED IN FEET / MINUTE = T.S.
SCALE OF P.A. ÷ D.A.
.00
.70
.60
.50
.40
.30
.20
.10
0
18.0
17.0
16.0
14.0
12.0
10.0
8.0
6.0
4.0
2.0
0
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
.10
.20
.30
.40
.50
.60
.70

TABLE I

SHEET 1 of 3

PROPELLER ANALYSIS AND DESIGN

BASIC CONDITIONS

$A = \frac{E.H.P \times R \times C'}{P \times 10{,}000}$ $A' = \frac{S.H.P \times R \times C'}{P \times 10{,}000}$ $B = \frac{E.H.P \times C}{P \times R \times D^2}$

VALUES OF C AND C'			
No. OF BLADES	2	3	4
C	398	291.8	252.4
C'	3.98	2.918	2.524

P.A / D.A.	$S.T_d$.	T.S. / T^2	T.S. / 318.35	T.S.×S.T.$_d$ / T^2	P.C.'	A 3 BLADES	A 4 BLADES	A'	B 3 BLADES	B 4 BLADES
.150	1.02	1406	13.87	1434	.763	1.54	1.54	2.02	.778	.778
.155	1.08	1434	14.14	1549	.764	1.70	1.70	2.22	.825	.825
.160	1.14	1456	14.37	1660	.766	1.85	1.85	2.42	.873	.873
.165	1.20	1483	14.64	1780	.766	2.02	2.02	2.64	.919	.919
.170	1.27	1506	14.86	1913	.767	2.21	2.21	2.88	.974	.974
.175	1.34	1537	15.17	2060	.767	2.43	2.43	3.17	1.028	1.028
.180	1.41	1560	15.39	2200	.768	2.64	2.62	3.43	1.083	1.095
.185	1.47	1585	15.64	2330	.768	2.84	2.82	3.69	1.129	1.142
.190	1.55	1607	15.86	2491	.769	3.09	3.05	4.01	1.192	1.190
.195	1.63	1630	16.08	2657	.769	3.33	3.28	4.33	1.253	1.240
.200	1.71	1660	16.38	2839	.769	3.62	3.56	4.71	1.315	1.296
.205	1.78	1679	16.57	2989	.769	3.86	3.85	5.02	1.369	1.353
.210	1.85	1703	16.81	3151	.769	4.12	4.15	5.36	1.423	1.410
.215	1.93	1728	17.06	3335	.769	4.43	4.44	5.76	1.484	1.467
.220	2.01	1754	17.31	3526	.768	4.75	4.74	6.18	1.544	1.525
.225	2.12	1782	17.59	3778	.768	5.18	5.03	6.74	1.628	1.582
.230	2.19	1802	17.78	3946	.768	5.46	5.35	7.11	1.682	1.632
.235	2.26	1824	18.00	4122	.766	5.76	5.67	7.52	1.731	1.682
.240	2.35	1849	18.25	4345	.765	6.14	5.99	8.03	1.798	1.732
.245	2.42	1875	18.50	4538	.763	6.48	6.63	8.50	1.846	1.782
.250	2.51	1904	18.79	4779	.762	6.93	6.64	9.10	1.913	1.832
.255	2.60	1926	19.00	5008	.761	7.34	7.05	9.65	1.976	1.888
.260	2.69	1948	19.22	5240	.760	7.74	7.46	10.21	2.039	1.944
.265	2.78	1974	19.48	5488	.758	8.20	7.87	10.83	2.108	1.999
.270	2.87	2002	19.76	5746	.757	8.71	8.28	11.51	2.173	2.055
.275	2.96	2028	20.01	6003	.755	9.19	8.69	12.18	2.235	2.111
.280	3.05	2053	20.26	6262	.753	9.67	9.18	12.85	2.300	2.168
.285	3.15	2075	20.48	6536	.751	10.19	9.67	13.55	2.366	2.224
.290	3.24	2101	20.73	6807	.750	10.74	10.15	14.31	2.430	2.281
.295	3.33	2126	20.98	7080	.748	11.27	10.64	15.05	2.491	2.337
.300	3.43	2154	21.25	7388	.746	11.88	11.13	15.92	2.559	2.394
.305	3.52	2180	21.52	7674	.743	12.44	11.69	16.73	2.615	2.445
.310	3.61	2206	21.77	7964	.741	13.03	12.26	17.58	2.675	2.496
.315	3.70	2228	21.99	8244	.739	13.59	12.82	18.37	2.734	2.548
.320	3.78	2260	22.30	8543	.737	14.25	13.39	19.31	2.786	2.599
.325	3.89	2292	22.62	8916	.734	15.00	13.95	20.42	2.855	2.650
.330	4.00	2317	22.87	9268	.732	15.74	14.64	21.48	2.928	2.707
.335	4.10	2346	23.15	9619	.729	16.46	15.32	22.57	2.989	2.765
.340	4.20	2375	23.43	9975	.727	17.21	16.01	23.68	3.053	2.822
.345	4.31	2403	23.72	10357	.725	18.05	16.69	24.90	3.125	2.880
.350	4.42	2432	24.00	10749	.723	18.91	17.38	26.15	3.196	2.937
.355	4.53	2457	24.25	11130	.720	19.70	18.16	27.33	3.262	2.991

TABLE I (CONTINUED) SHEET 2 OF 3

PROPELLER ANALYSIS AND DESIGN

BASIC CONDITIONS

P.A./D.A.	S.Td.	T.S./π	T.S./318.35	T.S.×S.Td./π	P.C.'	A		A'	B	
						3 BLADES	4 BLADES		3 BLADES	4 BLADES
·360	4·62	2483	24·50	11471	·718	20.48	18.94	28.50	3.320	3.045
·365	4·73	2515	24·82	11896	·715	21.38	19.71	29.90	3.300	3.098
·370	4·84	2543	25·10	12308	·713	22.31	20.49	31.30	3.450	3.152
·375	4·95	2569	25·35	12717	·710	23.30	21.27	32.67	3.515	3.206
·380	5·06	2592	25·63	13115	·708	24.10	22.15	34.00	3.585	3.263
·385	5·17	2626	25·91	13576	·705	25.14	23.03	35.63	3.644	3.320
·390	5·28	2652	26·17	14003	·703	26.11	23.95	37.15	3.649	3.376
·395	5·39	2680	26·45	14445	·700	27.10	24.79	38.70	3.775	3.433
·400	5·52	2711	26·75	14965	·698	28.35	25.67	40.60	3.855	3.490
·405	5·63	2737	27·01	15409	·696	29.37	26.79	42.17	3.92	3.544
·410	5·74	2769	27·33	15894	·693	30.50	27.90	44.00	3.98	3.598
·415	5·87	2801	27·64	16442	·690	31.80	29.02	46.05	4.05	3.652
·420	5·98	2833	27·96	16941	·688	33.05	30.13	48.00	4.115	3.706
·425	6·11	2865	28·27	17505	·685	34.80	31.25	51.80	4.185	3.760
·430	6·23	2897	28·58	18048	·683	35.70	32.38	52.30	4.25	3.816
·435	6·35	2928	28·90	18593	·681	37.05	33.51	54.40	4·33	3·872
·440	6·48	2960	29·21	19181	·678	38.50	34.64	56.75	4·395	3·928
·445	6·62	2992	29·53	19807	·676	40.05	35.77	59.25	4·475	3·984
·450	6·74	3024	29·84	20382	·673	41.45	36.90	61.60	4·537	4·040
·455	6·87	3059	30·19	21015	·670	43.07	38.35	64.25	4·600	4·094
·460	6·99	3094	30·53	21627	·668	44.70	39.80	66·40	4·67	4·148
·465	7·11	3129	30·88	22247	·665	46.30	41.25	69.60	4·73	4·202
·470	7·27	3164	31·22	23002	·663	48.25	42.70	72·80	4·82	4·256
·475	7·40	3199	31·57	23673	·660	49·15	44.15	75·65	4·885	4·310
·480	7·52	3234	31·91	24320	·658	51.90	45·85	78·65	4·950	4·359
·485	7·64	3279	32·35	25052	·655	53·80	47·55	82·10	5·000	4·409
·490	7·77	3317	32·73	25773	·653	55·80	49·25	85·40	5·07	4·458
·495	7·91	3358	33·14	26562	·650	58·00	50·95	89·20	5·14	4·508
·500	8·05	3400	33·55	27370	·648	60.30	52·65	93.00	5.22	4.557
·505	8·19	3428	33·83	28075	·646	62·15	54·56	96·20	5·29	4·607
·510	8·30	3476	34·30	28851	·643	64.45	56.47	100.3	5.335	4.657
·515	8·46	3511	34·65	29703	·641	66·90	58.38	104.4	5·42	4·708
·520	8·60	3549	35·02	30521	·638	69.10	60.29	108.4	5.49	4.758
·525	8·74	3547	35·50	31438	·636	71·95	62·20	113.1	5.56	4.808
·530	8·88	3635	35·87	32279	·633	74·25		117.4	5.62	
·535	9·02	3676	36·28	33157	·631	76.85		121.9	5·69	
·540	9·16	3724	36·75	34112	·628	79.80		127.1	5·755	
·545	9·31	3769	37·19	35089	·626	82·80		132.3	5·825	
·550	9·46	3820	37·69	36137	·623	86·10		138·1	5·89	
·555	9·61	3861	38·10	37104	·621	89·00		143·4	5·965	
·560	9·75	3909	38·57	38113	·618	92·10		149·1	6.025	
·565	9·90	3957	39·05	39174	·616	95.50		155.0	6·100	
·570	10·05	4008	39·55	40240	·613	99.00		161.5	6·16	
·575	10.21	4055	40.02	41402	·611	102·60		168.0	6·24	
·580	10·37	4106	40.52	42579	·608	106·30		174.8	6.305	

TABLE I (CONTINUED) SHEET 3 of 3

PROPELLER ANALYSIS AND DESIGN

BASIC CONDITIONS

P.A / D.A	$S.T_d$	T.S / π	T.S / 318.35	T.S x S.T_d / π	P('	A 3 BLADES	A 4 BLADES	A'	B 3 BLADES	B 4 BLADES
.585	10.52	4157	41.02	43732	.606	110		182	6.315	
.590	10.66	4208	41.53	44857	.604	114		189	6.44	
.595	10.81	4265	42.09	46105	.602	119		197	6.51	
.600	10.97	4310	42.53	47281	.594	122		204	6.57	
.605	11.12	4361	43.03	48494	.597	124		211	6.64	
.610	11.29	4402	43.44	49699	.595	130		219	6.715	
.615	11.44	4453	43.95	50942	.592	134		227	6.77	
.620	11.59	4501	44.42	52167	.589	139		235	6.825	
.625	11.77	4558	44.98	53648	.587	143		244	6.91	
.630	11.91	4606	45.45	54857	.584	148		253	6.955	
.635	12.07	4663	46.02	56282	.582	153		262	7.025	
.640	12.25	4717	46.55	57783	.580	158		273	7.105	
.645	12.42	4768	47.06	59219	.577	163		283	7.165	
.650	12.59	4837	47.73	60898	.575	170		295	7.240	
.655	12.74	4886	48.22	62248	.572	175		305	7.285	
.660	12.93	4940	48.75	63874	.569	180		316	7.355	
.665	13.09	4997	49.32	65411	.567	186		327	7.420	
.670	13.26	5061	49.95	67109	.565	192		340	7.495	
.675	13.43	5112	50.45	68654	.562	197		351	7.54	
.680	13.57	5166	50.98	70103	.559	202		362	7.585	
.685	13.75	5230	51.61	71913	.557	209		376	7.660	
.690	13.91	5284	52.14	73500	.554	215		388	7.705	
.695	14.09	5344	52.74	75297	.552	222		402	7.775	
.700	14.26	5411	53.40	77161	.550	229		417	7.845	

TABLE 1A

CORRECTION FACTORS FOR BUILT-UP PROPELLERS

DIAMETER OF HUB / DIAM. OF PROPELLER	.20	.21	.22	.23	.24	.25	.26	.27	.28	.29	.30	.31	.32
P.C. CORRECTION FACTOR	.993	.989	.985	.979	.974	.968	.962	.955	.948	.941	.934	.926	.917
R CORRECTION FACTOR			1.00	1.001	1.002	1.003	1.005	1.007	1.01	1.012	1.014	1.018	1.021

$R = \frac{T.S \div \pi}{D}$; $V/1-S_B = \frac{T.S}{318.35} \times \frac{P}{D}$; $S.H.P = \frac{T.S \times S.T_d}{\eta} \times \frac{D \times P}{C}$; $E.H.P = S.H.P. \times P.C.$

of the formulas, it is divided by π. For convenience in making calculations, values of $TS \div 318.35$ and $(TS \times ST_d) \div \pi$ have also been given. The values of A, A′ and B are used only in design calculations and will be explained later.

If two propellers having the same diameter, pitch and number of blades but with projected area ratios varying by .005 (such as .330 and .335 or .550 and .555) be analyzed for the actual shaft horsepower and revolutions, it will be found that the variations will be less than one-half of one per cent. With such small variations, it is not necessary to interpolate should the projected area of the propeller to be analyzed fall between any two values given in Table I. In such cases use the nearest value of P.A. ÷ D.A.

The values of P.A. ÷ D.A. given in the first column are the projected area ratios of 3 blades, that is the total projected area ratio of a 3-bladed propeller, 3/4 of the total projected area of 4-bladed propellers and 3/2 of the total of 2-bladed wheels, and should, in all cases, be used in obtaining the value of $S.T._d$, $T.S. \div \pi$, $T.S. \div 318.35$, $(T.S. \times ST_d) \div \pi$, A,A′ and B. The value of P.C.′, however, should be taken for the total projected area ratio (4/3 P.A. ÷ DA for 4-bladed and 2/3 P.A. ÷ D.A. for 2-bladed wheels).

Table 1A gives correction factors for basic propulsive coefficients and revolutions when built-up propellers are used.

NUMBER OF BLADES AND THEIR EFFECT ON POWER AND PROPULSIVE EFFICIENCY.

Chart 6 and Table I were developed from data of performances of 3-bladed propellers, and, therefore a correction must be applied in calculations for shaft horsepower and for obtaining the propulsive coefficients if it should be desired to use any other number of blades.

If three wheels having the same diameter, pitch, shape of blades, and the same area per blade, but having different number of blades be operating at the same revolutions and slip and under the same wake conditions, it will be found that the four-bladed propeller will absorb 15.6 per cent more power while the two-

bladed wheel will absorb only 75 per cent of the power required by the three-bladed wheel. The propulsive coefficients, however, will be greatest for the two-bladed and least for the four-bladed wheels.

FORMULAS FOR BASIC CONDITIONS.

As already shown the formula for indicated thrust of a 3-bladed propeller is:

$$S.T._d = (291.8 \times SHP) \div (P \times R \times D^2)$$

Therefore S.H.P. $= (ST_d \times P \times R \times D^2) \div 291.8$

substituting TS $\div \pi$ D for R we get

$$SHP = \frac{ST_d \times P \times TS \times D^2}{291.8 \times \pi \times D} = \frac{T.S. \times ST_d}{\pi} \times \frac{P \times D}{291.8}$$

$$\text{Now } V = \frac{P \times R \times (1 - S_B)}{101.33} \text{ Or } \frac{V}{1 - S_B} = \frac{P \times R}{101.33}$$

substituting TS $\div \pi$D for R we get

$$\frac{V}{1 - S_B} = \frac{P \times TS}{\pi D \times 101.33} = \frac{TS}{318.35} \times \frac{P}{D}$$

Having given the diameter (D), pitch (P) and the projected area ratio (P.A. $\div$ D.A.), the values of (T.S. $\times$ S.T.$_d$) $\div \pi$ and TS $\div$ 318.35 are obtained from Table I. The basic shaft horsepower (S.H.P.), and the basic speed divided by one minus basic slip ($V/1\text{-}S_B$) are obtained by the use of the above formulae.

Obtain from Table I the value of P.C.′ If a built-up propeller is used, obtain from Table 1A the correction factor for the ratio of the diameter of hub to the diameter of propeller (d $\div$ D). The corrected propulsive coefficient (P.C.) will then be

$$P.C. = P.C.' \times \text{correction factor from Table 1A}$$

and the basic effective horsepower will be

$$E.H.P. = S.H.P. \times P.C.$$

Obtain from Table I the value of $TS \div \pi$. The basic revolution per minute will be

$$\text{Basic Rpm.} = R = (TS \div \pi) \div D$$

If a built-up propeller is used it will be necessary to apply a correction factor (R.C.F.) if the diameter of the hub is greater than .26 of the diameter of the propeller. This correction factor is found in Table 1A. The formula for basic revolutions now becomes

$$R = \frac{TS \div \pi}{D} \times \text{R.C.F.}$$

When dealing with propellers of reduced diameter (fan-shaped blades) the actual diameter of the propeller (D_1) and not the diameter of the basic standard propeller (D) should be used in obtaining the values of $d \div D$. for correction factors for both P.C. and R.

The above formula for basic shaft horsepower (S.H.P.) is for a 3-bladed propeller, therefore, if a 2- or 4-bladed wheel is to be used it will be necessary to apply a correction as already explained.

For a 4-bladed wheel the formula for basic shaft horsepower becomes

$$\text{S.H.P.} = \frac{TS \times ST_d}{\pi} \times \frac{P \times D \times 1.156}{291.8} = \frac{TS \times ST_d}{\pi} \times \frac{P \times D}{252.4}$$

and for a 2 bladed propeller.

$$\text{S.H.P.} = \frac{TS \times ST_d}{\pi} \times \frac{P \times D \times .75}{291.8} = \frac{TS \times ST_d}{\pi} \times \frac{P \times D}{398}$$

In both cases the total projected area ratio should be used for obtaining the value of P.C.'

FORM FOR OBTAINING THE BASIC CONDITIONS.

The form PROPELLER ANALYSIS — BASIC CONDITIONS give the various steps necessary to obtain the basic revolutions (R line 8), basic speed divided by one minus basic slip ($V/1\text{-}S_B$ line 10), basic shaft horsepower (S.H.P. line 13) and the basic effective horsepower (E.H.P. line 17).

PROPELLER ANALYSIS
BASIC CONDITIONS

TYPE OF PROPELLER - SOLID OR BUILT UP — DIAMETER OF HUB

RATIO: DIAMETER OF HUB ÷ DIAMETER OF PROPELLER

NUMBER OF BLADES — NUMBER OF PROPELLERS

TYPE OF BLADE SECTION

E	D_1 = ACTUAL PROPELLER DIAM. IN FEET
F	$D \div D_1$ = RATIO: BASIC DIAM. ÷ ACTUAL DIAM.
H	$\sqrt{D \div D_1}$
J	$\sqrt[3]{D \div D_1}$
1	N = N° OF PROPELLERS
2	D = BASIC DIAM. OF PROPELLER IN FEET
3	P = PITCH IN FEET
4	PA ÷ D.A. = PROJECTED AREA RATIO OF 3 BLADES
5	4/3 PA. ÷ D.A. = TOTAL PROJ. AREA RATIO OF 4 BLADES
6	T.S ÷ π (TABLE I) *
7	R.C.F = REVOLUTION CORRECTION FACTOR (TABLE IA) X
8	R = [(T.S ÷ π) × √(D ÷ D_1) × R.C.F ÷ D = [(6) × H × (7)] ÷ (2) = BASIC R.P.M
9	T.S ÷ 318.35 (TABLE I)*
10	$V/1-S_B = \frac{T.S.}{318.35} \times \frac{P}{D} = (9) \times \frac{(3)}{(2)}$
11	T.S × $S.T_d$. ÷ π (TABLE I)*
12	(N × D × P) ÷ C = [(1) × (2) × (3)] ÷ C
	3 BLADED WHEELS C = 291.0 : 4 BLADED WHEELS C = 252.4 : 2 BLADED WHEELS C = 348
13	$N \times S.H.P = \frac{T.S \times S.T_d}{\pi}$ TOTAL BASIC SHAFT HORSE-POWER ON ALL SHAFTS $\times \frac{N \times D \times P}{C} = (11) \times (12)$
14	P.C' = BASIC PROPULSIVE COEFF. (TABLE I)†
15A	P.C CORRECTION FACTOR FOR BLADE SECTION
15B	P.C " " " ROUGHNESS
15C	P.C " " " HUB DIAM. (TABLE IA) X
15D	TOTAL CORR. FACTOR = [(15A) × (15B) × (15C)] ÷ H or J
16	P.C = P.C' × TOTAL CORRECTION FACTOR = (14) × (15D)
17	N × E.H.P = N × S.H.P × P.C = (13) × (16) = BASIC EFFECTIVE H.P
18	$1-S_B$ = 1 - BASIC SLIP - FROM HULL ANALYSIS
19	K POWER AUGMENT FACTOR - FROM HULL ANALYSIS

NOTE: LINES E.F & H ARE FOR PROPELLERS OF REDUCED DIAM. (FAN SHAPE BLADES). PROPELLERS WITH STANDARD FORM OF PROJ. AREA THE ACTUAL DIAM (D_1) = THE BASIC DIAM. (D). FOR THESE PROPELLERS LINES E, F & H ARE OMITTED

* IN ALL CASES (3 AND 4 BLADED WHEELS) USE PROJ. AREA RATIO OF 3 BLADES (PA ÷ D.A. LINE 4) FOR OBTAINING T.S ÷ π, T.S. ÷ 318.35 AND T.S × $S.T_d$. ÷ π FROM TABLE I

† FOR OBTAINING P.C' FROM TABLE I USE THE FOLLOWING
3 BLADED WHEELS USE PA ÷ D.A. LINE 4
4 " " " 4/3 PA ÷ DA LINE 5

X FOR BUILT-UP WHEELS ONLY

When dealing with propellers of standard forms of projected areas lines E, F, H and J are omitted. These are only used where it is necessary to widen the propeller blade tips, when it is impossible to use the diameter that the calculations show as necessary. The use of these lines will be explained later.

The propulsive coefficients given on Chart 6 and Table I are for solid propellers with standard shape projected areas and with ogival blade section (Section A, Fig. 6). If a built-up propeller is used, a correction as already explained should be made. Should the shape of the projected area and/or the blade section differ materially from the standard, corrections should be made in accordance with instructions in Part 4.

METHOD FOR CHANGING FROM BASIC CONDITIONS TO OTHER CONDITIONS OF RESISTANCE AND SPEED.

When a vessel is operating under conditions other than the basic, the shaft horsepower and revolutions will change. The change in shaft horsepower will depend upon the change in effective horsepower, but will not vary with a change in speed from that of the basic conditions provided the propeller is not operating in cavitation. The revolutions, however, will change with a change in the shaft horsepower and/or speed.

Correction for Shaft Horsepower.—Although the shaft horsepower varies with a change of effective horsepower it will not vary in direct ratio.

Letting E.H.P. = basic effective horsepower

e.h.p. = actual effective horsepower

S.H.P. = basic shaft horsepower

$S.H.P._a$ = actual shaft horsepower

Z = net power correction factor depending upon e.h.p. ÷ E.H.P. (See Table II)

PROPELLER ANALYSIS AND DESIGN

Sheet N°1 of 3

TABLE II

e.h.p. / E.H.P.	Values of Z									
	0	1	2	3	4	5	6	7	8	9
.01	.0083	.0092	.0100	.0109	.0118	.0127	.0135	.0144	.0153	.0161
.02	.0170	.0179	.0188	.0197	.0206	.0215	.0224	.0233	.0242	.0251
.03	.0260	.0269	.0278	.0287	.0296	.0305	.0314	.0323	.0332	.0341
.04	.0350	.0359	.0368	.0378	.0387	.0396	.0405	.0414	.0424	.0433
.05	.0442	.0451	.0460	.0470	.0479	.0488	.0497	.0506	.0516	.0525
.06	.0534	.0543	.0553	.0562	.0571	.0581	.0590	.0599	.0608	.0618
.07	.0627	.0636	.0646	.0655	.0665	.0674	.0683	.0693	.0702	.0712
.08	.0721	.0730	.0740	.0749	.0759	.0768	.0777	.0787	.0796	.0806
.09	.0815	.0824	.0834	.0843	.0853	.0862	.0871	.0881	.0890	.0900
.10	.0909	.092	.093	.094	.095	.096	.097	.098	.099	.100
.11	.101	.101	102	103	.104	.105	.106	.107	.108	.109
.12	.110	.111	.112	.113	.114	.115	.116	.117	.118	.119
.13	.120	.121	.122	.123	.124	.125	.126	.126	.127	.128
.14	.129	.130	.131	.132	.133	.134	.135	.136	.137	.138
.15	.139	.140	.141	.142	.143	.144	.145	.146	.147	.148
.16	149	.150	.151	.152	.152	.153	.154	.155	.156	.157
.17	.158	.159	.160	.161	.162	.163	164	.165	.166	.167
.18	.168	.169	.170	.171	.172	.173	.174	.175	.176	.177
.19	.178	.178	.179	.180	.181	.182	.183	.184	.185	.186
.20	.187	.188	.189	.190	.191	.192	.193	.194	195	.196
.21	.197	.198	.199	.200	.201	.202	.203	.204	.205	.206
.22	.207	.208	.209	.210	211	.212	.213	.214	.215	.216
.23	.217	.218	.219	.220	.221	.221	.222	.223	.224	.225
.24	.226	.227	.228	.229	.230	.231	.232	.233	.234	.235
.25	.236	.237	.238	.239	.240	.241	.242	.243	.244	.245
.26	.246	.247	.248	.249	.250	.251	.252	.253	.254	.255
.27	.256	.257	.258	.259	.260	.261	.262	.263	.264	.265
.28	.266	.267	.268	.269	.270	.271	.272	.273	.274	.275
.29	.276	.277	.278	.279	.280	.281	.281	.282	.283	.284
.30	.285	.286	.287	.288	.289	.290	.291	.292	.293	.294
.31	.295	.296	.297	.298	.299	.300	.301	.302	.303	.304
.32	.305	.306	.307	.308	.309	.310	.311	.312	.313	.314
.33	.315	.316	.317	.318	.319	.320	.321	.322	.323	.324
.34	.325	.326	.327	.328	.329	.330	.331	.332	.333	.334
.35	.335	.336	.337	.338	.339	.340	.341	.342	.343	.344
.36	.345	.346	.347	.348	.349	.350	.351	.352	.353	.354
.37	.355	.356	.357	.358	.359	.360	.361	.362	.363	.364
.38	.365	.366	.367	.368	.369	.370	.371	.372	.373	.374
.39	.375	.376	.377	.378	.379	.380	.381	.382	.383	.384
.40	.385	.386	.387	.388	.389	.390	.391	.392	.393	.394

PROPELLER ANALYSIS AND DESIGN

SHEET No 2 OF 3

TABLE II CONTINUED

e.h.p. / E.H.P.	VALUES OF Z									
	0	1	2	3	4	5	6	7	8	9
.41	.395	.396	.397	.398	.399	.400	.401	.402	.403	.404
.42	.405	.406	.407	.408	.409	.410	.411	.412	.413	.414
.43	.415	.416	.417	.418	.419	.420	.421	.422	.423	.424
.44	.425	.426	.427	.428	.429	.430	.431	.432	.433	.434
.45	.436	.437	.438	.439	.440	.441	.442	.443	.444	.445
.46	.446	.447	.448	.449	.450	.451	.452	.453	.454	.455
.47	.456	.457	.458	.459	.460	.461	.462	.463	.464	.465
.48	.466	.467	.468	.469	.470	.471	.472	.473	.474	.475
.49	.476	.477	.478	.479	.480	.481	.482	.483	.484	.485
.50	.486	.487	.488	.489	.490	.491	.492	.493	.494	.495
.51	.496	.497	.498	.499	.500	.501	.502	.503	.504	.505
.52	.506	.507	.508	.509	.510	.511	.512	.513	.514	.515
.53	.516	.517	.518	.519	.520	.521	.523	.524	.525	.526
.54	.527	.528	.529	.530	.531	.532	.533	.534	.535	.536
.55	.537	.538	.539	.540	.541	.542	.543	.544	.545	.546
.56	.547	.548	.549	.550	.551	.552	.553	.554	.555	.556
.57	.557	.558	.559	.560	.561	.562	.563	.564	.565	.566
.58	.567	.568	.569	.570	.571	.572	.573	.574	.575	.576
.59	.577	.578	.579	.580	.581	.582	.584	.585	.586	.587
.60	.588	.589	.590	.591	.592	.593	.594	.595	.596	.597
.61	.598	.599	.600	.601	.602	.603	.604	.605	.606	.607
.62	.608	.609	.610	.611	.612	.613	.614	.615	.616	.617
.63	.618	.619	.620	.621	.622	.623	.624	.625	.626	.627
.64	.628	.629	.630	.631	.632	.633	.635	.636	.637	.638
.65	.639	.640	.641	.642	.643	.644	.645	.646	.647	.648
.66	.649	.650	.651	.652	.653	.654	.655	.656	.657	.658
.67	.659	.660	661	662	.663	.664	.665	.666	.667	.668
.68	.669	670	.671	.672	.673	.674	.676	.677	.678	.679
.69	.680	.681	.682	.683	.684	.685	.686	687	.688	.689
.70	.690	.691	.692	.693	.694	.695	.696	.697	.698	.699
.71	.700	.701	.702	.703	.704	.705	.706	.707	.708	.709
.72	.710	.711	.712	.713	.714	.715	.716	.717	.718	.719
.73	.721	.722	.723	.724	.725	.726	.727	.728	.729	.730
.74	.731	.732	.733	.734	.735	.736	.737	.738	.739	.740
.75	.741	.742	.743	.744	.745	.746	.747	.748	.750	.751
.76	.752	.753	.754	.755	.756	.757	.758	.759	.760	.761
.77	.762	.763	.764	.765	.766	.767	.768	.769	.770	.771
.78	.772	.773	.774	.775	.776	.777	.778	.779	.780	.781
.79	.782	.783	.784	.785	.786	.787	.789	.790	.791	.792
.80	.793	.794	.795	.796	.797	.798	.799	.800	.801	.802

PROPELLER ANALYSIS AND DESIGN Sheet No 3 of 3

TABLE II Continued

e.h.p. / E.H.P.	Values of "Z"									
	0	1	2	3	4	5	6	7	8	9
.81	.803	.804	.805	.806	.807	.808	.809	.810	.811	.812
.82	.813	.814	.815	.816	.817	.818	.819	.820	.821	.822
.83	.824	.825	.826	.827	.828	.829	.830	.831	.832	.833
.84	.834	.835	.836	.837	.838	.839	.840	.841	.842	.843
.85	.844	.845	.846	.847	.848	.849	.850	.852	.853	.854
.86	.855	.856	.857	.858	.859	.860	.861	.862	.863	.864
.87	.865	.866	.867	.868	.869	.870	.871	.872	.873	.874
.88	.875	.876	.877	.878	.879	.880	.882	.883	.884	.885
.89	.886	.887	.888	.889	.890	.891	.892	.893	.894	.895
.90	.896	.897	.898	.899	.900	.901	.902	.903	.904	.905
.91	.906	.907	.908	.910	.911	.912	.913	.914	.915	.916
.92	.917	.918	.919	.920	.921	.922	.923	.924	.925	.926
.93	.927	.928	.929	.930	.931	.932	.933	.934	.935	.936
.94	.938	.939	.940	.941	.942	.943	.944	.945	.946	.947
.95	.948	.949	.950	.951	.952	.953	.954	.955	.956	.957
.96	.958	.959	.960	.961	.963	.964	.965	.966	.967	.968
.97	.969	.970	.971	.972	.973	.974	.975	.976	.977	.978
.98	.979	.980	.981	.982	.983	.984	.985	.986	.988	.989
.99	.990	.991	.992	.993	.994	.995	.996	.997	.998	.999
1.0	1.000	1.010	1.021	1.031	1.042	1.052	1.062	1.073	1.083	1.094
1.1	1.104	1.115	1.125	1.136	1.146	1.156	1.167	1.178	1.188	1.199
1.2	1.209	1.220	1.230	1.241	1.251	1.262	1.272	1.283	1.293	1.304
1.3	1.314	1.325	1.335	1.346	1.356	1.367	1.378	1.388	1.400	1.410
1.4	1.420	1.431	1.441	1.452	1.462	1.473	1.483	1.494	1.505	1.515
1.5	1.525	1.536	1.547	1.557	1.568	1.578	1.589	1.599	1.611	1.621
1.6	1.632	1.643	1.653	1.664	1.674	1.685	1.696	1.706	1.717	1.727
1.7	1.738	1.748	1.759	1.770	1.780	1.791	1.803	1.812	1.823	1.833
1.8	1.844	1.854	1.865	1.876	1.887	1.898	1.908	1.919	1.930	1.940
1.9	1.951	1.962	1.972	1.983	1.994	2.004	2.015	2.026	2.037	2.048
2.0	2.058									

and assuming that cavitation does not exist, and the power augment factor K is unity we have.

$$S.H.P._d = S.H.P. \times Z$$

If, however, the power augment factor K is other than unity, the formula for actual shaft horsepower becomes

$$S.H.P._d = S.H.P. \times Z \times K$$

Letting $Z \times K = Z_g$ (the gross power factor) we have

$$S.H.P._d = S.H.P. \times Z_g$$

When the power augment factor K is unity, the gross power factor (Z_g) is equal to the net power factor (Z).

Correction for Revolutions per Minute.—As explained earlier, the revolutions will vary with a change in shaft horsepower and/or speed.

Letting V = basic speed in knots = $(V/1\text{-}S_B) \times (1\text{-}S_B)$

v = actual speed in knots

Z_r = revolution factor depending upon the gross load factor Z_g [$S.H.P._d \div$ S.H.P. (Chart 8A).]

Y = revolution factor depending upon the speed fraction $v \div V$ (Chart 8A)

R = basic revolutions per minute

R_d = actual revolutions per minute

The actual revolutions are obtained by the following formula:

$$R_d = R \times Z_r \times Y$$

It should be noticed that the values of the correction factor Y as shown on Chart 8A is different for 3- and 4-bladed propellers when the speed fraction ($v \div V$) exceeds 1.00. The question has been asked "Why this difference between 3- and 4-bladed wheels?" The only answer is that analyses of obtainable data shows this to be the case.

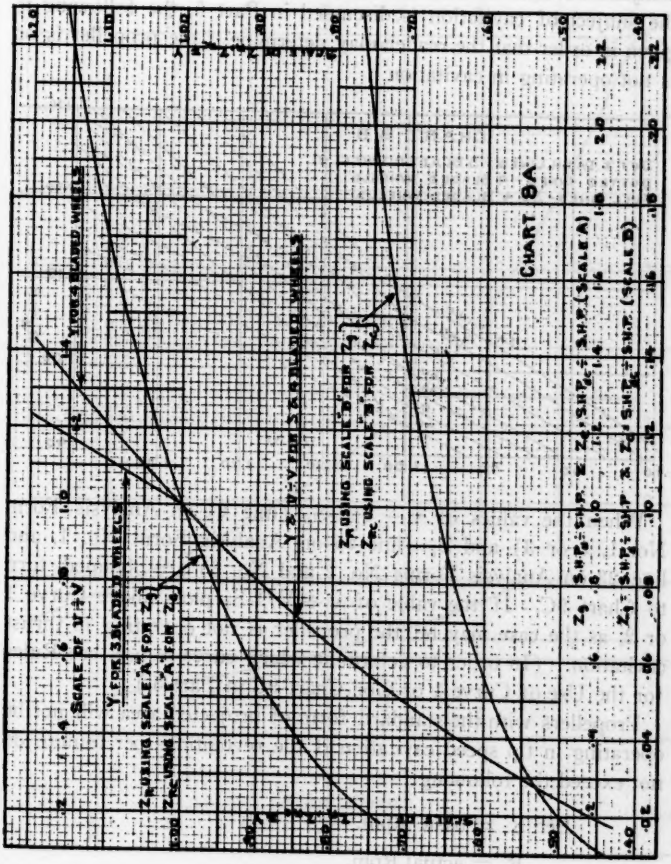
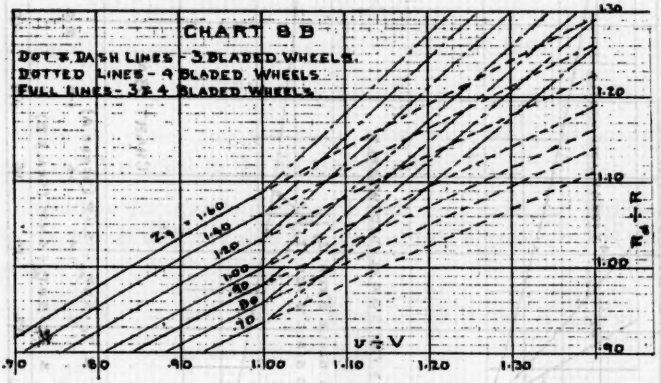


CHART 8A
Y FOR 3 BLADED WHEELS
Y = V FOR 3 & 4 BLADED WHEELS
TWIN SCREW BOARD WHEELS
SCALE OF V ÷ √A
Z_R USING SCALE 'A' FOR Z_A
Z_R USING SCALE 'B' FOR Z_A
Z_R USING SCALE 'A' FOR Z_A
Z_R USING SCALE 'B' FOR Z_A
Z_A = S.H.P. ÷ S.H.P. R. (SCALE A)
Z_A = S.H.P. ÷ S.H.P. R. (SCALE B)

ANALYSIS FORMS.

Analysis Forms 1, 2 and 3 can be used for making propeller analyses when the screw is not operating in cavitation. Although the subject of cavitation is discussed in Part 5, the following is given in order that the reader may determine if the propeller is or is not operating in cavitation.



When the values of $v \div V/1\text{-}S_B$ (line 21 on Analysis Form No. 1, 2 or 3) and Z_g (line 25 on Analysis Form No. 1, and line 23 on Analysis Forms No. 2 and No. 3) are obtained, turn to Chart 9C. If the value of $v \div V/1\text{-}S_B$ plots above Curve 1 or 2, as the case may be, the propeller is not operating in thrust cavitation. (In order to determine which curve to use, see Rules for the Use of Curve 1 or 2 on Chart 9C in Part 5.)

Propellers with the standard form of projected area will not be operating in tip speed cavitation if the revolutions per minute do not exceed the following:

Let R = basic Rpm.

R_d = actual Rpm.

Propellers with ogival sections (Section A of Figure 6)

3-bladed propellers $R_d = R$

4-bladed propellers $R_d = .95\ R$

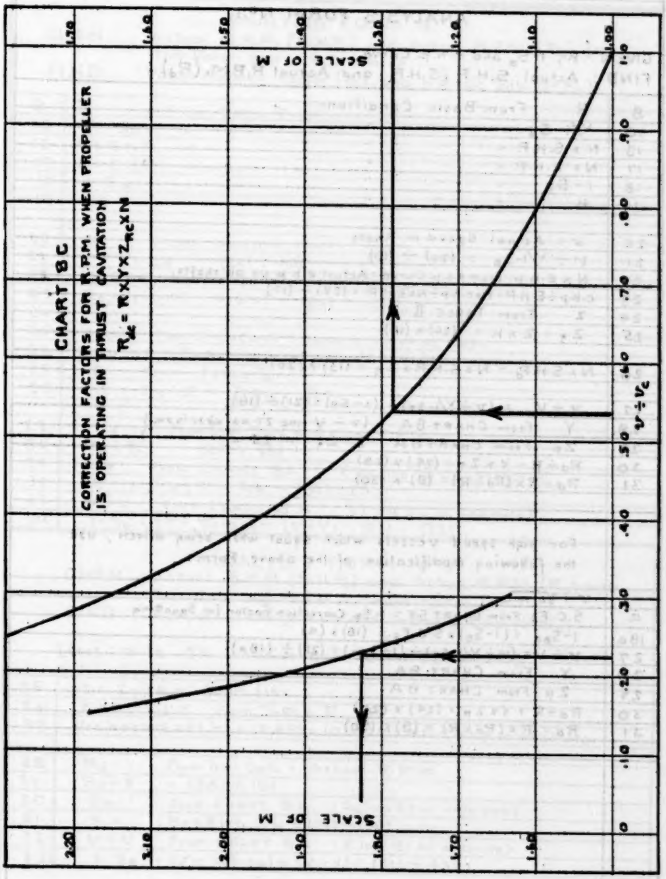


CHART 8C
CORRECTION FACTORS FOR R.P.M. WHEN PROPELLER IS OPERATING IN THRUST CAVITATION
$T_{ac} = R \times Y \times Z_{RCN}$
SCALE OF M
SCALE OF M
V ÷ Vc

PROPELLER ANALYSIS
NOT IN CAVITATION

ANALYSIS FORM N° 1

GIVEN: K, $1-S_B$ and e.h.p. Curve
FIND: Actual S.H.P. ($S.H.P_d$) and Actual R.P.M. (R_d)

8	R From Basic Conditions
10	$V/1-S_B$ " " "
13	N x S.H.P. " " "
17	N x E.H.P. " " "
18	$1-S_B$ " " "
19	K " " "
20	v = Actual Speed in Knots
21	$v \div V/1-S_B$ = (20) ÷ (10)
22	N x e.h.p. From e.h.p. Curve = Actual e.h.p. on all shafts.
23	e.h.p ÷ E.H.P = N x e.h.p ÷ N x E.H.P = (22) ÷ (17)
24	z From TABLE II
25	$Z_q = Z \times K$ = (24) x (19)
26	$N \times S.H.P_d = N \times S.H.P. \times Z_q$ = (13) x (25)
27	$v \div V$ = $(v \div V/1-S_B) \div (1-S_B)$ = (21) ÷ (18)
28	Y From CHART 8A ($v \div V$ line 27 as abscissa)
29	Z_R From CHART 8A (Z_q " 25 " ")
30	$R_d \div R = Y \times Z_R$ = (29) x (28)
31	$R_d = R \times (R_d \div R)$ = (8) x (30)
	For high speed vessels which squat when being driven, use the following modification of the above Form.
	$v \div \sqrt{L.W.L}$
a	S.C.F_S From CHART 5F = $1-S_B$ Correction Factor for Squatting
18a	$1-S_{Ba}$ = $(1-S_B) \times$ S.C.F_S = (18) x (a)
27	$v \div V$ = $(v \div V/1-S_a) \div (1-S_{Ba})$ = (21) ÷ (18a)
28	Y From CHART 8A
29	Z_R From CHART 8A
30	$R_d \div R = Y \times Z_R$ = (29) x (28)
31	$R_d = R \times (R_d \div R)$ = (8) x (30)

PROPELLER ANALYSIS
NOT IN CAVITATION

ANALYSIS FORM N° 2

GIVEN : Actual S.H.P. ($S.H.P_d$) and Actual R.P.M. (R_d) from Trial Data and e.h.p curve

FIND : Power Augment Factor (K) and 1-Basic Slip ($1-S_B$)

8	R	From Basic Conditions
10	$V/1-S_B$	" " "
13	N x S.H.P	" " "
17	N x E.H.P.	" " "
18	$1-S_B$	" " "
19	K	" " "
20	v = Actual Speed in Knots	
21	$v \div V/1-S_B$:	(20) ÷ (10)
22	N x $S.H.P_d$	From Trial Data Actual S.H.P on all shafts
23	Z_q = N x $S.H.P_d \div$ N x S.H.P =	(22) ÷ (13)
24		
25	N x e.h.p	From e.h.p. Curve = Actual e.h.p on all shafts
26	e.h.p. ÷ E.H.P. = N x e.h.p ÷ N x E.H.P :	(25) ÷ (17)
27	Z	From TABLE II
28	$K = Z_q \div Z$ =	(23) ÷ (27)
29	R_d	From Trial Data = Actual R.P.M.
30	$R_d \div R$	(29) ÷ (8)
31	Z_R	From CHART 8A (Z_q line 23 as abscissa)
32	$Y = (R_d \div R) \div Z_R$ =	(30) ÷ (31)
33	$v \div V$	From CHART 8 A (Y line 32 as ordinate)
34	$1-S_B = (v \div V/1-S_B) \div (v \div V)$:	(21) ÷ (33)

ANALYSIS FORM N°3

GIVEN : Actual S.H.P. ($S.H.P_d$) and Actual R.P.M. (R_d) from Trial Data and Power Augment Factor (K)

FIND : Actual E.H.P (e.h.p.) and 1-Basic Slip ($1-S_B$)

Lines 7 to 24 inclusive same as in Form N°. 2

25	$Z = Z_q \div K$ =	(23) ÷ (19)
26	e.h.p ÷ E.H.P.	From TABLE II
27	N x e.h.p = N x E.H.P x (e.h.p ÷ E.H.P) =	(17) x (26) = Actual e.h.p. on all shafts
28	R_d	From Trial Data = Actual R.P.M.
29	$R_d \div R$	= (28) ÷ (8)
30	Z_R	From CHART 8A (Z_q line 23 as abscissa)
31	Y =	$(R_d \div R) \div Z_R$ = (29) ÷ (30)
32	$v \div V$	From CHART 8A (Y line 31 as ordinate)
33	$1-S_B = (v \div V/1-S_B) \div (v \div V)$:	(21) ÷ (32)

Propellers with section similar to Section G of Figure 6

3-bladed propellers $R_d = 1.15\ R$

4-bladed propellers $R_d = 1.09\ R$

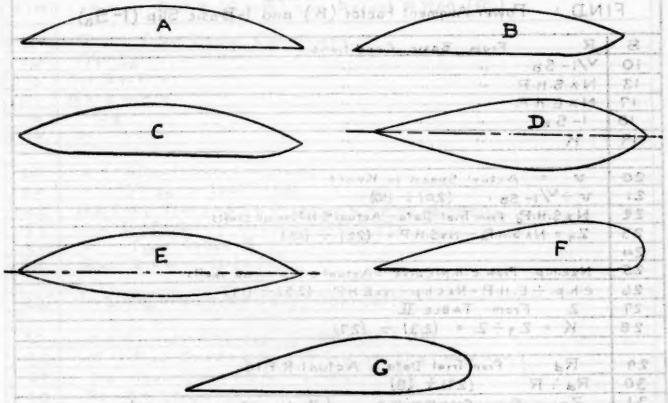


FIGURE 6.

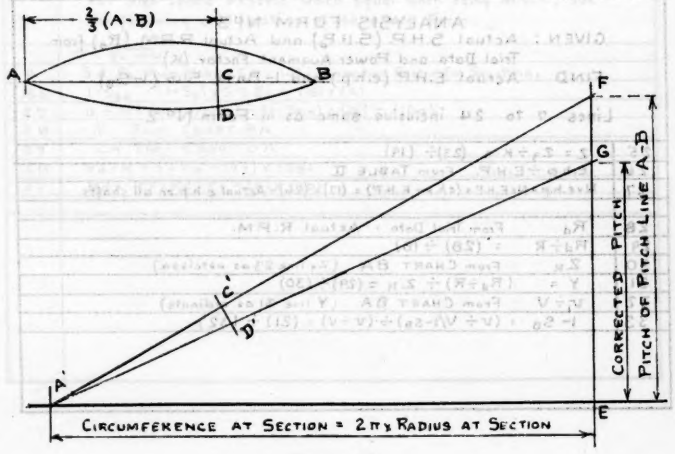


FIGURE 6A.

Analysis Form No. 1.—This form is used when it is desired to ascertain the shaft horsepower and revolutions required for a given speed. In such cases it is necessary to have the effective horsepower curve derived from tank tests, or if this is lacking, an estimated curve, the propeller's characteristics and the hull data so that the values of K and $1\text{-}S_B$ can be determined.

Analysis Form No. 2.—This form is used when the characteristics of the propeller, trial data and the effective horsepower curves are available, it being desired to find the power augment factor K and the value of $1\text{-}S_B$.

Analysis Form No. 3.—This form is used when it is required to obtain the effective horsepower and the value of $1\text{-}S_B$ from trial data. Under such conditions, it is necessary to have the propeller characteristics, and for obtaining the effective horsepower it is necessary to know the value of K.

ACCURACY TO BE EXPECTED.

In making estimates of the expected performances of propellers on trial or in service, considerable differences between the estimated and the actual performances may, at times be expected. These differences are caused by the difference in trial conditions of the ship and the model.

Conditions under which the model is towed and the curve of effective horsepower obtained are as nearly ideal as possible; the wetted surface of the model being in the best of condition as to smoothness; the water in the tank smooth and quiet; the air still; and finally, the model is constrained to move in a perfectly straight line.

The conditions under which the ship is tried may include the following:

(*a*) The wetted surface of the hull may be rougher in proportion than that of the model, producing increase frictional resistance.

(*b*) Improper design or careless fitting of submerged appendages to the hull, producing increased resistance.

(*c*) The trim may be such as to cause a change in wake, a change in the value of the power augment factor K, or cause

heavy eddying around the appendanges. These will cause changes in the power and revolutions.

(*d*) Excessive engine losses due to improper adjustment and lubrication.

(*e*) Excess roughness of the propeller surface or excessive bluntness of the following edges of the blades.

(*f*) Incorrect setting of the pitch or incorrect pitch data.

(*g*) Unfavorable weather and sea conditions.

(*h*) Existence of cross and eddy currents on the trial course.

(*i*) Shallow water on the trial course.

(*k*) Erratic steering on the trial course.

(*l*) Errors of instruments for measuring power and revolutions.

(*m*) Unavoidable errors of observations due to the personal equation of the observers.

(*n*) Carelessness of the observers and personnel in running trials.

With the above list of handicaps, it should be considered satisfactory if the predicted performance agrees reasonably close to the actual, especially in cases where the actual power exceeds the estimated.

When making estimates of the expected performance of vessels, any error in K will result in the same percentage error in $S.H.P._d$, but will not affect the revolutions, except in so far as the revolutions are affected by a change in power, which in most cases will be very small.

If the trial results are to be analyzed for the effective horsepower delivered by the propeller, an error in K will make an error in the e.h.p. which will be very nearly, but not exactly, the same per cent as the error in K.

An error in $1\text{-}S_B$ will affect the revolutions but not the power. The error in Rpm. will be about one-half that in $1\text{-}S_B$, or a four per cent error in $1\text{-}S_B$ will make a two per cent error in revolutions.

SHIP HELD STATIONARY.

When a ship is held stationary, as during a dock trial, the revolutions can be approximated by the use of Chart 8D. Although the ratio of the depth of water to the draft of the ship will affect the revolutions somewhat, it will be found that the use of this chart will in most cases give quite satisfactory results.

THRUST.

In order to provide a proper size thrust bearing it is necessary to determine the thrust. This will be considerably greater than the resistance, due to the action of the propeller which draws the water away from the hull causing an increase in resistance. As this is only one of the factors which determine the value of K, this value cannot be used directly, but can be used as a basis to determine the ratio of thrust to resistance.

Letting e.h.p = actual effective horsepower

K = power augment factor

Res. = resistance in pounds

T = thrust in pounds

v = speed in knots

We have $\text{Res.} = (325.8 \times \text{e.h.p.}) \div \text{v}$

and the ratio of thrust to resistance will be

$\text{T} \div \text{Res.} = .875\text{K} + .395$ for single screw vessels

$\text{T} \div \text{Res.} = .54\text{K} + .625$ for twin screw vessels

These formulas have been checked with data of a number of self-propelled tests and thrust measurements on ships, where the latter were available. In the case of single screw ships the greatest differences were 8 per cent over and 7 per cent under, and with twin screw vessels 5 per cent over and 7 per cent under.

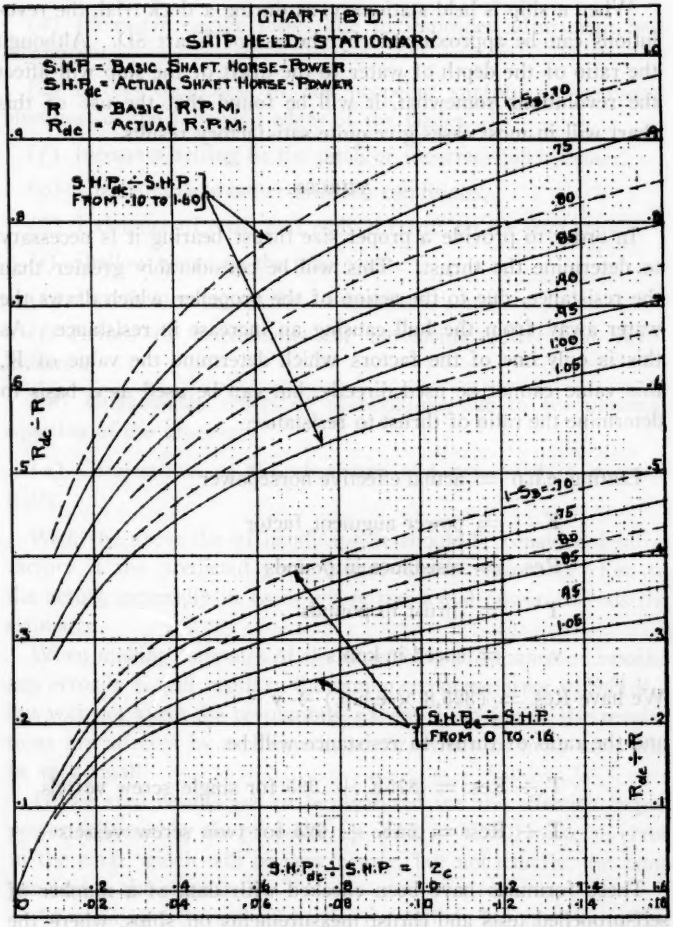


CHART 8 D
SHIP HELD STATIONARY
S.H.P = Basic Shaft Horse-Power
S.H.P$_{dc}$ = Actual Shaft Horse-Power
R = Basic R.P.M.
R$_{dc}$ = Actual R.P.M.
S.H.P$_{dc}$ ÷ S.H.P From .10 to 1.60
1-S$_s$=.10
.75
.80
.85
.90
.95
1.00
1.05
R$_{dc}$ ÷ R
S.H.P$_{dc}$ ÷ S.H.P From 0 to .16
S.H.P$_{dc}$ ÷ S.H.P = Z$_c$

Part 4—Form of Projected Area and Blade Section, Constant and Varying Pitch, Number of Blades and Rake of Blades, Condition of Blade Surface.

FORM OF PROJECTED AREA.

The Dyson Standard.—In order to maintain a constant distribution of blade area and thus guard against changes in resistance due to changes in distribution, Admiral Dyson adopted as a basic form of projected area the form of projection of the blades of two 3-bladed propellers having approximately a projected area ratio of .32. These two propellers had excellent records, which could hardly be bettered.

Using a standard hub diameter equal to .2 of the diameter of the propeller, this standard basic form of projection was drawn. With the center of the hub as a center, circular arcs were drawn at different radii crossing the projection. In obtaining the projected form of areas differing from the original area the widths of the projections measured on these circular arcs were made proportional to the circular arc widths of the original projections; that is a .6 projected area ratio would have circular arc measurements 60/32 times as great as those of the original .32 projection.

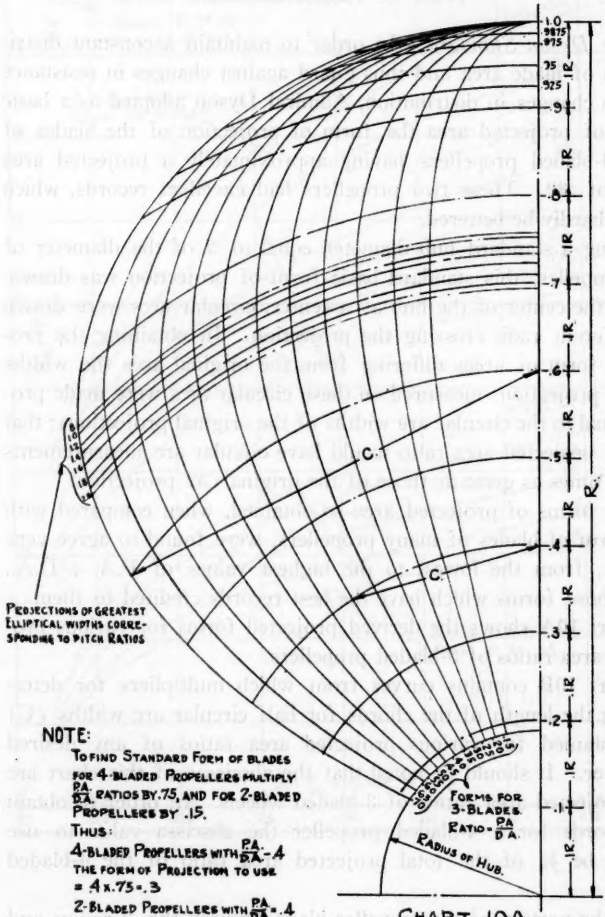
The forms of projected area so obtained, when compared with the form of blades of many propellers, were found to agree very closely, from the lowest to the highest values of P.A. ÷ D.A., with those forms which have the best records credited to them.

Chart 10A shows the derived projected forms for various projected area ratios of 3-bladed propellers.

Chart 10B contains curves from which multipliers for determining the length of the chords for half circular arc widths (C) are obtained for various projected area ratios of any desired diameter. It should be noted that the abscissae of this chart are for projected area ratios of 3-bladed wheels. In order to obtain the chords for a 4-bladed propeller the abscissa value to use should be ¾ of the total projected area ratio of the 4-bladed wheel.

As the portion of the propeller blade between the .7 radius and the hub appears to have but slight effect upon the performance of

DYSON STANDARD PROJECTED AREA FORMS FOR SCREW PROPELLERS WITH 3-4 AND 2 BLADES.



NOTE:

TO FIND STANDARD FORM OF BLADES FOR 4-BLADED PROPELLERS MULTIPLY $\frac{PA}{DA}$ RATIOS BY .75, AND FOR 2-BLADED PROPELLERS BY .15.

THUS:

4-BLADED PROPELLERS WITH $\frac{PA}{DA} = .4$ THE FORM OF PROJECTION TO USE $= 4 \times .75 = .3$

2-BLADED PROPELLERS WITH $\frac{PA}{DA} = .4$ THE FORM OF PROJECTION TO USE $= .4 \times 1.5 = .6$

CHART 10A

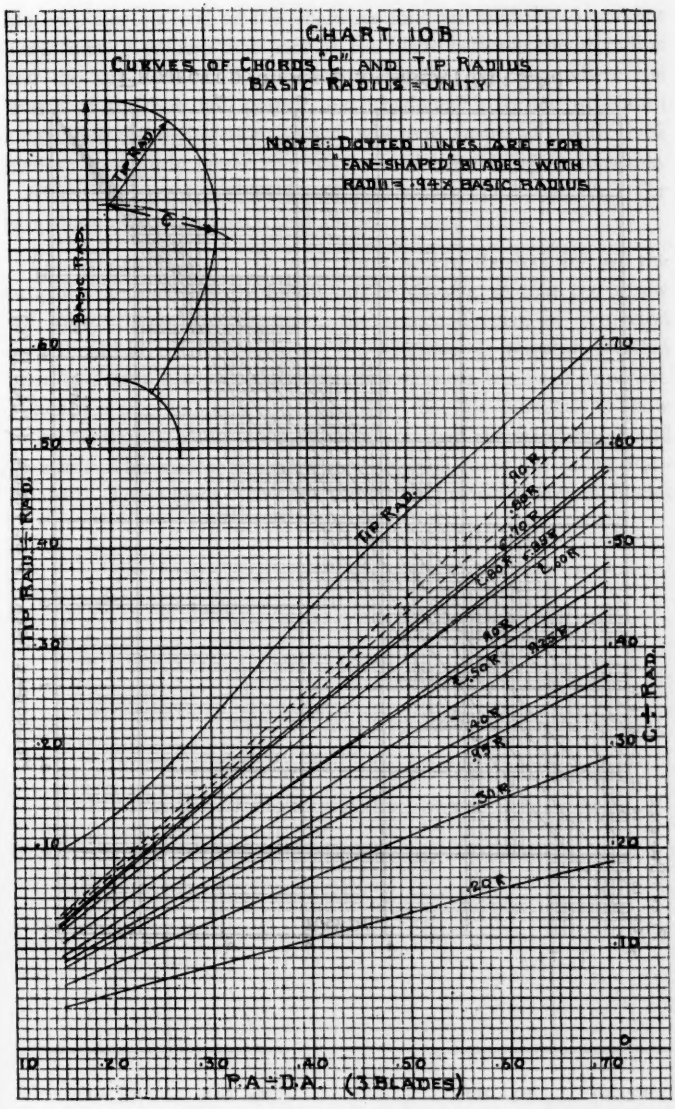
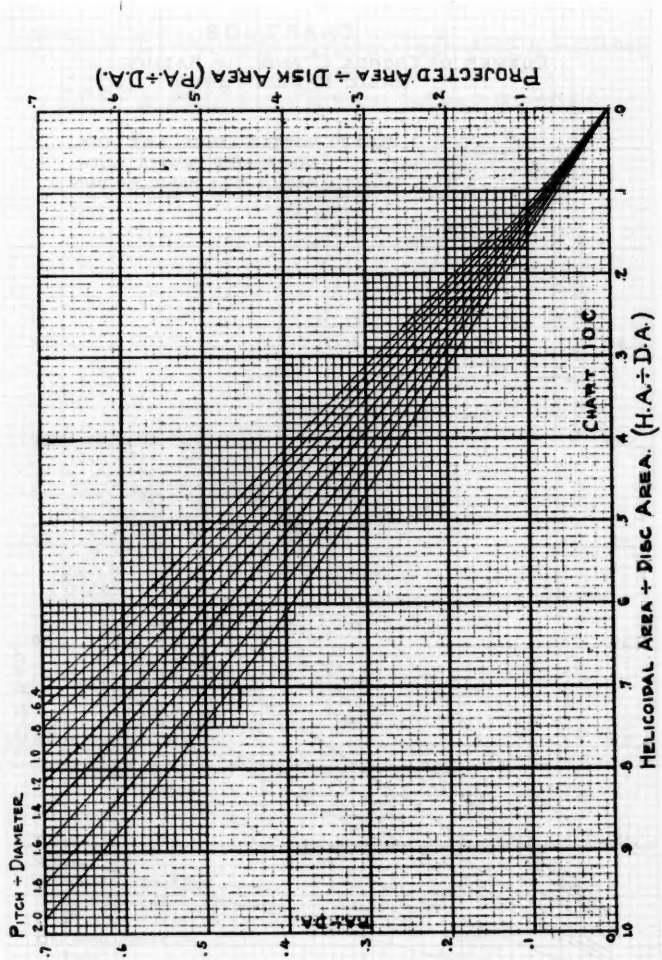


CHART 10B
CURVES OF CHORDS "C" AND TIP RADIUS
BASIC RADIUS = UNITY
NOTE: DOTTED LINES ARE FOR "FAN-SHAPED" BLADES WITH RADII = .94 X BASIC RADIUS
TIP RAD
.60
.50
.40
.30
.20
.10
.70
.80
.90
1.00
.50
.40
.30
.20
.10
0
.10
.20
.30
.40
.50
.60
.70
P.A.=D.A. (3 BLADES)


PROJECTED AREA ÷ DISK AREA (PA÷DA.)
PITCH ÷ DIAMETER.
HELICOIDAL AREA ÷ DISC AREA. (H.A.÷D.A.)
CHART 10C
HA÷DA

the propeller so long as the blade sections are such as to prevent losses due to eddies, that part of the blade between the .7 radius and the hub need not be adhered to rigidly, but may, where the length of the hub will permit, be broadened in order to obtain longer and thinner sections, which will tend (especially with thick blades) to slightly better the efficiency if any change is possible.

Chart 10C gives the developed area ratios (H.A. ÷ D.A.) for propellers of various pitch and projected area ratios.

Fan Shape Blades.—In new work where the location of the propeller and/or the revolution can be chosen, so that the screw will have the proper diameter, the standard form of projected area should be used. However, the Engineer is sometimes called upon to design a wheel where insufficient room has been provided. When it is impossible to use a diameter that the calculations show to be necessary, blades with broader tips than the standard should be used. The blades generally used in such cases are called Fan-Shaped. These blades have the same total projected area as blades

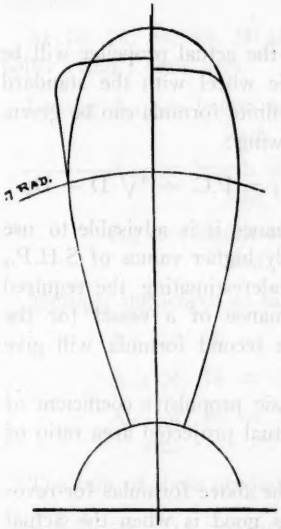


FIG. 4A FIG. 4B

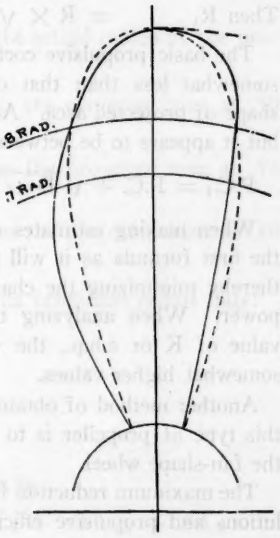


FIG. 5A FIG. 5B

of standard form whose diameter is greater than that of the actual propeller, the amount of surface of the basic wheel outside the circumference of the actual propeller being added to the sides from the .7 radius out, in order to maintain the same area as the basic screw. Such a blade is shown in Figure 4A.

The propulsive coefficient and revolutions of propellers with fan-shaped blades will vary from the standard.

Letting D = diameter of the basic propeller

D_1 = diameter of the actual propeller

R = basic Rpm. of the basic propeller

R_1 = basic Rpm. of the actual propeller

P.C. = basic propulsive coefficient of the basic propeller from Table I

$P.C._1$ = basic propulsive coefficient of the actual propeller ·

Then $R_1 = R \times \sqrt{D \div D_1}$

The basic propulsive coefficient of the actual propeller will be somewhat less than that of the basic wheel with the standard shape of projected area. As yet no definite formula can be given, but it appears to be between the following:

$$P.C._1 = P.C. \div \sqrt{D \div D_1} \text{ or } P.C._1 = P.C. \div \sqrt[3]{D \div D_1}$$

When making estimates of performance it is advisable to use the first formula as it will give slightly higher values of $S.H.P._d$ thereby minimizing the chances of underestimating the required power. When analyzing the performance of a vessel for the value of K or e.h.p., the use of the second formula will give somewhat higher values.

Another method of obtaining the basic propulsive coefficient of this type of propeller is to use the actual projected area ratio of the fan-shape wheel.

The maximum reduction for which the above formulas for revolutions and propulsive efficiency holds good is when the actual diameter is .90 of the basic diameter ($D \div D_1 = 1.11$). When

the leading and following edges of the blade from the hub to the tip are straight or concave lines and the radius of the corners small use 1.11 as the value of $D \div D_1$ and a projected area equal to that of the actual propeller. The projected area ratio of the basic propeller will then be:

$$P.A. \div D.A. = (P.A.' \div D.A.') \div (D \div D_1)^2$$

$P.A. \div D.A.$ = projected area ratio of the basic propeller

$P.A.' \div D.A.'$ = projected area ratio of the actual propeller

D = diameter of the basic propeller

D_1 = diameter of the actual propeller

The pitch of the basic propeller will be the same as that of the actual propeller.

Broad Tip Blades.—When the tip of the basic blade is cut off as shown in Figure 4B the blades are called Broad Tip. When it is required to analyze propellers with blades of this type, the equivalent fan-shaped blade should be obtained, which is done as follows:

At .70, .75, .80, .85, .90 and .95 of the actual radius of the propeller (Rad.) draw arcs.

Letting $A_{.7}$ = length of the arc on the projected area at .70 Rad.

$A_{.75}$ = length of the arc on the projected area at .75 Rad.

$A_{.8}$ = length of the arc on the projected area at .80 Rad., etc.

Multiply the length of each arc by its fractional radius thus:

$$A_{.7} \times .70 = .70A_{.7}$$

$$A_{.75} \times .75 = .75A_{.75}$$

$$A_{.8} \times .80 = .80A_{.8}$$

etc.

The sum of these products will then be

$$\Sigma \text{ R.A.} = .7A_{.7} + .75A_{.75} + .8A_{.8} + .85A_{.85} + .9A_{.9} + .95A_{.95} + A_{1.0}$$

Let $B = \Sigma$ R.A. $\div$ .7A.$_7$

D = diameter of the basic propeller with standard form of projected area

D_1 = actual diameter of the propeller

Then $D \div D_1 = .026B + .834$

and $D = D_1 \times (D \div D_1)$

The projected area ratio of the basic propeller can, in most cases, be obtained with sufficient accuracy for practical purposes by the following formula:

$$P.A. \div D.A. = (P.A.' \div D.A.') \div (D \div D_1)^2$$

P.A. $\div$ D.A. = projected area ratio of the basic propeller

P.A.$' \div$ D.A.$'$ = projected area ratio of the actual propeller

D and D_1 are the same as above

The pitch for the basic propeller will be the same as the actual pitch.

If the projected area ratio of one blade is less than .12 the length of the chords of half the arc may be taken in place of the arcs.

Narrow Tip Blades.—When the greatest width of the projected circular arc is at a radial distance from the center of the propeller considerably less than the standard form (.7 Radius), the blades are known as Narrow Tipped, an example of which is shown in Figure 5A. In order to obtain the equivalent projected area ratio of such blades proceed as follows:

(*a*) Draw an arc at .8 of the actual radius of the propeller and divide this arc in halves.

(*b*) Obtain the length of the chord of one-half of the arc and divide this by the radius of the propeller (C $\div$ Rad.)

(*c*) Obtain from Chart 10B the abscissa value of curve .8 Rad. where its ordinate value is equal to C $\div$ Rad. This value will be the equivalent standard projected area ratio of 3 blades.

The propeller can then be analyzed as a standard wheel with a diameter and pitch equal to those of the actual wheel and a projected area ratio as found above.

As these blades have a larger actual projected area ratio than the equivalent standard, they will enter tip speed cavitation at slightly higher revolutions than their equivalent standard, but will enter thrust cavitation at about the same time.

Bulbous Tip Blades.—Blades having more area outside the .7 radius than the standard, but whose tips do not coincide with the circumference of the propeller, as shown in Figure 5B, are known as Bulbous Tip Blades. When the shape of the blade from the .7 radius to the tip does not differ materially from the standard, it may be treated as a standard shape in the same manner as narrow tip blades. If, however, there should be a considerable difference, such wheels should be treated in the same manner as those with Broad Tip Blades.

Odd Shape Blades.—All of the above blade shapes are met in disguised form by throwing the blade in one direction or the other, and yet their radial distribution by this distortion may not be changed. When dealing with such blades proceed as follows:

(*a*) Draw arcs at the .7, .8, .85, .9 and .95 radii and divide same into halves.

(*b*) Obtain the length of the chords of half of the arcs and their ratio to the radius of the propeller (C ÷ Rad.).

(*c*) By comparing the length of the chords with those of a standard shape with a circular width at the .7 Rad. equal to that of the blade in question, it can be determined which type of blade shape should be used.

BLADE SECTIONS.

The principal forms of blade sections encountered in practice are shown in Figure 6.

Section A.—This is what is known as the ogival section. The faces of the blades with this section form the nominal pitch surfaces, the thickness being built on the back with a form of an arc of a circle. This section was used on the propellers from which

performances the values of tip speed (T.S.), indicated thrust (ST_d) and propulsive coefficient (P.C.') shown by the curves on Chart 6 (also values in Table I) were derived, and therefore is considered as the standard section, although it is not the section that gives the best performance.

Section B.—This section is similar to Section A, but has the leading edge thrown back. It was claimed by the advocates of this section that the efficiency was somewhat better than wheels with the ogival section (Section A). Although this claim was disputed, recent self-propelled model tests and trials of vessels indicate that a saving in power can be obtained by throwing back the leading edge.

Section C.—This section is similar to Section B, but has the following as well as the leading edge thrown back. A modification of this section is one where only the following edge is thrown back. Throwing back the following edge is a mistake, as the tendency of the water to break contact with the blade face is aggravated, and the formation of eddies at this edge, with consequent vibration and loss of efficiency is invited.

Sections D and E.—Blades of Section D with the metal divided evenly on each side of the nominal helical pitch plane, offer less resistance to turning than any of the other sections. When such sections are used the revolutions will be greater than with blades of Sections A or B. This section is shown with the thickest part nearer the leading than the following edge, but it may be midway between the two. A modification of this section is when the metal is not evenly divided on each side of the nominal helical pitch as in Section E. When the driving face is only slightly convex the difference in revolutions and propulsive efficiency from those of Section A will probably be within the error of observation.

Section F.—This is known as the air-foil section. The face of the blade conforms with the helix while the back is shaped similar to an airplane wing, the leading edge being round and the thickest part of the blade being about one-third of its width from the leading edge. At one time considerable gain in efficiency was claimed for this section, which claim, however, was not substantiated by self-propelled model tests or in service. Self-propelled

model tests have shown, in some cases, no differences in propulsive efficiencies between blades with sections of this shape and those of Section A, the diameter, pitch, projected area, the number of blades and shape of projected area being the same for both propellers. In other cases, there was a gain of about 2 per cent at low powers but no gain at the design power. This section, however, has one advantage in that the rounding of the leading edge reduces erosion on the blade's driving face.

Section G.—This section is similar to Section F except the leading edge is thrown back. Wheels with this shape of section have higher efficiencies and enter tip speed cavitation at higher revolutions than those with sections similar to A (the diameter, pitch, projected area ratio and the shape of the projected area and the number of blades being the same in both cases). By rounding and throwing back the leading edge, erosion on the driving face is reduced considerably, and in some cases this trouble has been eliminated by the use of this section.

The gain in efficiency will depend upon the throw back (or lift) of the leading edge. When the throw back of the leading edge at the .20 radius is .40 of the greatest thickness at that section, the basic propulsive coefficient is approximately 8 per cent greater than that of similar wheels with sections similar to A; that is, if a wheel having sections similar to Section A has a basic propulsive coefficient of .60, a propeller with the same number of blades, diameter, pitch, projected area ratio and shape of projected area, but with sections similar to Section F, will have a basic propulsive coefficient of .648.

At one time it was thought that throwing back the leading edge would have the same effect on the revolutions as slightly increasing the pitch, but it appears that such is not the case. The throw back of the leading edge very seldom extends beyond the .6 or .7 radius and therefore is in the part of the blade that is not very effective; furthermore, the back of such blades have less curvature (assuming the thickness is the same in both cases) which would tend to counteract any effective increase of the face pitch.

Efficiencies equal to those obtained by the use of Section G can be had by using a modification of this section. Such modifications

consist of blades with the leading edge rounded and thrown back (as in Section G), but the back of the blades are formed with arcs of circles, the thickest part of the blade near the hub (at or near the .2 radius) being from .33 to .40 of its width from the leading edge. As the throw back of the leading edge is advantageous only at sections near the hub, it should be decreased, becoming zero at the .6 or .7 radius. If there is any advantage in placing the thickest part of the blade nearer the leading edge than the following, it is only for the inner sections, therefore the distance of the greatest thickness from this edge may be increased for the outer sections until the .6 or .7 radius is reached. From this radius to the tip, the thickest part of the blade should be midway between the leading and the following edges.

Corrections for Convex Driving Faces.—When the driving face is convex, as shown in Sections D and E, the revolutions will be greater than when this face conforms with the helix. In order to correct for this difference, the pitch to be used in calculations for power, and to determine if the wheel is or is not operating in cavitation, as well as for revolutions, should be obtained as shown on Figure 6A.

The upper part of this Figure shows a blade section with a convex driving face. A-B is the width of the blade measured on the pitch line (which conforms with the helix). The distance A-C is equal to two-thirds (2/3) of A-B; C-D is the distance of the driving face from the pitch line at C.

Lay off A′-E as abscissa equal to the circumference of the circle at the radius of the section and E-F equal to the pitch of the pitch line A-B. Draw line A′-F. Lay off A′-C′ equal to A-C. Lay off C′-D′ perpendicular to A′-F and equal to C-D. Draw A′-D′ and extend same to G. E-G will be the pitch to be used at this radius. Follow this procedure for other sections. Should the corrected pitch at the various radii vary, the propeller should be treated as one with a varying pitch.

The scale for pitch (E-F and E-G) should be the same as that of the circumference (A′-E), and the scale of C′-D′ should be the same as that of A′-C′. It is not, however, necessary to have the scale of A-C′ and C′-D′ the same as that of A′-E and E-F.

CONSTANT AND VARYING PITCHES.

Constant Pitch Propellers.—The pitch of these propellers are the same throughout, except for a short distance from the leading edge when this edge is thrown back, as shown in sections B and G of Figure 6. Such wheels are easy to manufacture and the driving face can be readily machined when it is desired to do so.

Varying Pitch Propellers.—In almost all cases the pitch of these propellers increase from the hub to the tip. Extravagant claims have been made of the advantages of such propellers, but neither self-propelled model tests nor ship's trials have substantiated such claims. These wheels have the following disadvantages: First, they are more difficult to make. Second, it is harder to machine the blade surface, and if the increase in pitch is not constant such machining is impossible. Third, the cost of such propellers is greater than those with constant pitch.

When varying pitch propellers are to be analyzed, it is necessary to obtain what might be called the equivalent constant pitch, which is done as follows:

Let P_1 = average pitch from .7 radius to tip

P_2 = average pitch from .7 radius to .2 radius

P = equivalent constant pitch

Then $P = (2P_1 + P_2) \div 3$

If the difference in pitch is constant, that is the difference between the .3 and .4 radii is the same as that between the .4 and .5 radii and between the .5 and .6 radii, etc., the pitch at the .7 radius will be very close to that given by the above formula.

NUMBER OF BLADES.

The choice of the number of blades will depend upon conditions under which the propeller operates.

Two-bladed wheels are the most efficient, but are never used except on auxiliary sailing vessels and, in some cases, on small motor boats, as they are apt to cause vibrations. On auxiliary sailing vessels they have the advantage in that when sailing the blades can be placed in a vertical position behind the stern post, thus causing the least drag.

Three-bladed wheels are generally used on twin and quadruple screw vessels.

All sea-going single screw vessels should be fitted with four-bladed propellers. Although these wheels are not as efficient as those with three blades, they have a decided advantage should a blade be lost at sea.

Some people advocate the use of four-bladed on twin and quadruple screw vessels on the ground that they cause less vibration. This, however, is only true under certain cases. If the propellers are well submerged and there is sufficient clearance between the propeller tip and the hull and between the leading edge of the blade and strut, three-bladed propellers will operate as smoothly as those with four blades. This has been demonstrated on numerous occasions. If, however, the propellers are not well submerged or there is insufficient clearance between the propeller and the hull or strut, four-bladed wheels may reduce but will not eliminate the vibration.

If two wheels plot on the same point on Chart 9C and are operating under the same conditions of wake and resistance, the four-bladed propeller, although requiring more power, will have a slightly smaller diameter.

RAKE OF BLADES.

In some cases blades have been given a rake aft on the idea that centrifugal action of the screw on the water is decreased and the propulsive efficiency increased.

Examination of performances of actual propellers in service under certain conditions, and of model propellers in the tank, shows that there is no solid ground for either belief. In the case of full-size propellers, no differences in the propulsive efficiencies of propellers with and without rake can be noticed, while the model screws, with and without rake give practically identical results.

However, an actual advantage can, under certain conditions, be gained by raking the blades aft; for instance, greater tip clearances would be gained for wing screws, while greater distance between the leading edges of the blades and the after edges of the

struts or stern post would be furnished, thus tending to reduce the probability of encountering vibration when these clearances are small.

Where the tip and fore-and-aft clearances are large, radial blades are preferable to those with a rake as, in addition to being as efficient as the latter, they are more easily machined, have less total developed blade areas for equal projected areas, have less surface friction and are stiffer and lighter. The stresses in such blades due to centrifugal action are less than those of raked blades, a point which is of great importance for propellers of high tip speeds. For such propellers, the blades should never rake, but should be radial and symmetrical on each side of the central radial axis of the blades.

CONDITION OF BLADE SURFACE.

The condition of the blade surface has a very great effect on the propeller's efficiency, causing in some cases as much as a 10 per cent increase in power required for a given speed.

The S.S. *Clairton* was given three thorough tests, results of which were published in the Transaction of the Society of Naval Architects & Marine Engineers. The first two tests were run with an old propeller with a very rough surface, the third test was with a new wheel. After making allowances for wind resistance (see Transaction of the Society of Naval Architects & Marine Engineers, Vol. 40, 1932—"Trial Analysis Methods" by Lieut. Commander A. S. Pitre, C.C., U.S.N.), the power required for a given speed with the new wheel was about 10 per cent less than that with the old one.

The P.C. correction factor for roughness can be nothing more than a guess, although data on this subject is a considerable help in making this allowance. A well made new cast iron or steel propeller with smooth surfaces may have a correction factor as high as .96 to .98, but after a short time in service this will fall materially.

Although bronze wheels do not deteriorate as quickly as those made of cast iron or steel, they do become rough and therefore it is advisable to polish the blades at least every year or two.

The final portion of this article will be published in the February 1943 issue of the JOURNAL.

DISCUSSION.

The article to which the following commentary pertains was reprinted from the Brown-Boveri Review in the August 1942 issue of the JOURNAL. *Since this discussion is so important in connection with Dr. Meldahl's article it has been decided to make it available to readers of the* JOURNAL. *Difficulties of present day communications make it impracticable to communicate with Switzerland in time to permit this discussion to be printed by the original publisher. Dr. Meldahl will be invited to make a reply, if he desires, as early as circumstances permit.*

COMMENTARY ON A. MELDAHL'S PAPER "END LOSSES OF TURBINE BLADES."

BY D. ENG. J. KREITNER.

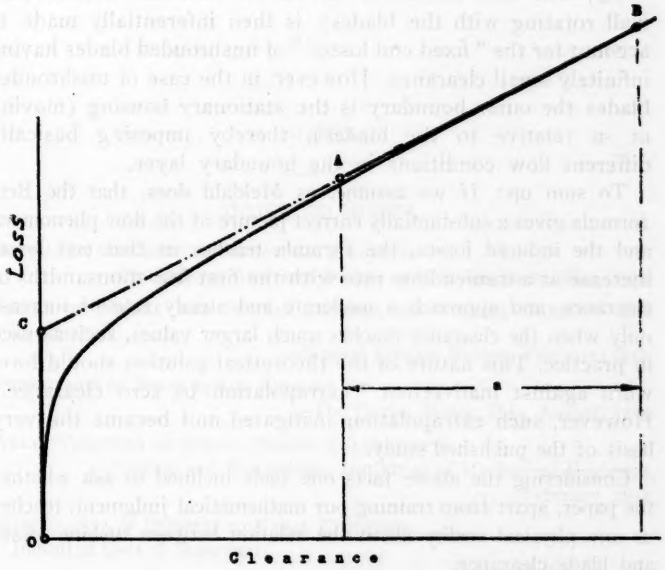
A. Meldahl's deductions appear intriguing at first glance. On scrutiny, one begins to wonder whether the reasoning is not somewhat similar to that of the man who told of lifting himself out of the marsh by grabbing his own hair.

This seems to be the paper's skeleton:

(A) It starts from a theoretical development by Betz, which involves an elliptic integral function such as shown in the attached sketch.

(B) The function is approximated by the straight line AB, which is of quite sufficient accuracy within the range "a" occurring in practice.

(C) In arithmetically interpreting the approximation formula thus obtained, the straight line is tacitly extrapolated to C, and the quantity OC presented as "fixed end loss, independent of the clearance s"; (formula (6), page 457).



(D) Various conversions of the approximation formula are discussed, all on the above basis of trying to attribute physical significance to the result of an improper arithmetical extrapolation.

(E) Test results are interpreted in a similar way; (page 456, line 10-12, and page 465): A substantially linear extrapolation of efficiencies measured at various finite clearances leads to values between .84 and .885 at zero clearance, while the pure profile efficiency had been measured at .91. The difference between these two values is taken as the above mentioned "fixed end loss." Actually it is, at least to a greater part, once more the result of the fallacious linearity in the extrapolation of the tests.

(F) The possibility of end losses without clearance is interestingly explained by discussing and demonstrating the secondary flow in the boundary layer near the ends of shrouded blades; (Fig. 2-6). No attempt is made to judge the magnitude of such losses.

(G) The flow mechanism shown for shrouded blades, (end wall rotating with the blades), is then inferentially made to account for the " fixed end losses " of unshrouded blades having infinitely small clearance. However, in the case of unshrouded blades the outer boundary is the stationary housing (moving at -u relative to the blades), thereby imposing basically different flow conditions in the boundary layer.

To sum up: If we assume, as Meldahl does, that the Betz formula gives a substantially correct picture of the flow phenomena and the induced losses, the formula teaches us that end losses increase at a tremendous rate with the first few thousandths of clearance, and approach a moderate and steady rate of increase only when the clearance reaches much larger values, such as used in practice. This nature of the theoretical solution should forewarn against inadvertent "extrapolation to zero clearance." However, such extrapolation instigated and became the very basis of the published study.

Considering the above facts one feels inclined to ask whether the paper, apart from training our mathematical judgment, teaches us any physical reality about the relation between turbine losses and blade clearance.

NOTES.

INDEX TO NOTES.

MECHANICS OF CREEP FOR STRUCTURAL ANALYSIS.
—Proceedings, American Society of Civil Engineers, May, 1942.

EFFECT OF NOZZLE AND BUCKET DEPOSITS ON TURBINE CAPACITY AND EFFICIENCY.
—General Electric Review, February, 1942.

SUPPRESSION OF NOISE IN OIL ENGINES.
—The British Motor Ship, August, 1942.

AXIAL VIBRATION OF DIESEL ENGINE CRANKSHAFTS.
—Journal and Proceedings, Institution of Mechanical Engineers, February, 1942.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

Industrial Uses of Stonewear.
—Engineering (London), May 1, 1942.

Rapid Scrap Identification Tests.
—Metals and Alloys, August, 1942.

Welding Fumes from Arc Welding.
—Journal, American Welding Society, June, 1942.

New Tube-Ice Machine.
—Transactions, Institute of Marine Engineers, April, 1942.

Angles in Space.
—Production Engineering, July, 1942.

Angles Between Planes.
—Production Engineering, August, 1942.

MECHANICS OF CREEP FOR STRUCTURAL ANALYSIS.

The continuous plastic deformation of constantly stressed materials under certain conditions is called "creep." The importance of creep in a particular design depends upon various factors such as the material considered, the allowable stress used, the temperature of operation, and the desirability of maintaining a small deformation. Creep sometimes assumes such magnitude in high-temperature designs that the necessity of considering creep effects has been realized for several years. On the other hand, little has been done to determine the influence of creep at normal temperatures as a factor in structural design. Its importance has been magnified by the recent interest shown in designing structural parts stressed to the plastic range. This paper by Joseph Marin, Professor of Engineering Mechanics at Pennsylvania State College, was published in the Proceedings of the American Society of Civil Engineers for May, 1942.

INTRODUCTION.

When they are subjected to stress, engineering materials sometimes develop plastic deformations with time, called "creep deformations." At high temperatures these deformations are of such magnitude that they definitely influence design. Considerable test data have been obtained at high temperatures, particularly for materials used in the turbine and oil-refining industries. (See, for example, "Compilation of Available High Temperature Creep Characteristics of Metals and Alloys," A. S. M. E.—A. S. T. M., 1938.) Many attempts have been made to interpret these data for the purpose of selecting working stresses.[2] To the structural engineer, the behavior of materials at normal temperatures is of greater interest, but for such temperatures there is little information available. A brief review of the information on creep in simple tension and at normal temperatures will be made first. This study leads to the selection of a new creep-stress law in place of Hooke's law for the development of a mechanics of creep.

NOTATION.

The letter symbols used in this paper are defined where they first appear and are assembled for reference in the Appendix.

SIMPLE TENSION; CREEP-STRESS LAW.

Usually the simple creep-tension test is used for obtaining basic data on the creep behavior of engineering materials. In this test a specimen is subjected to a stress that remains constant, and the creep deformations are measured with time. With this information creep-time relations can be plotted for each stress value. Figure 1 shows such plots for Aluminum Alloy 3S-3/4H tested at room temperature. These experiments in tension and others in pure bending have been reported in detail.[3] For many problems in design where creep is significant, it is necessary to select a working stress based on an allowable creep deformation in the estimated life of the machine or structural member. Industry cannot wait for lifetime test results, and it becomes necessary, therefore, to extrapolate test data that will apply for lengths of time

[2] "Interpretation and Use of Creep Results," by J. J. Kanter, Transactions, A. S. M., Vol. 24, No. 4, December, 1936, p. 870; see also "A Comparison of the Methods Used for Interpreting Creep Test Data," by J. Marin, Proceedings, A. S. T. M., Vol. 37, 1937, Pt. II, Technical Papers, p. 258.

[3] "Creep of Aluminum Subjected to Bending at Normal Temperatures," by J. Marin and L. E. Zwissler, Proceedings, A. S. T. M., Vol. 40, 1940, p. 937.

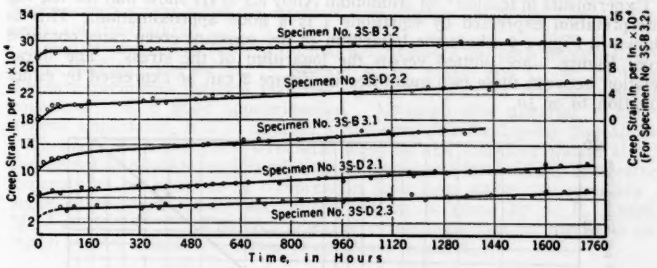

Specimen No. 3S-B 3.2
Specimen No 3S-D 2.2
Specimen No. 3S-B 3.1
Specimen No. 3S-D 2.1
Specimen No. 3S-D 2.3
Creep Strain, In. per In. x10⁴
Creep Strain, In per In. x10⁴ (For Specimen No 3S-B 3.2)
Time, in Hours
0 160 320 480 640 800 960 1120 1280 1440 1600 1760
2 6 10 14 18 22 26 30 34
0 4 8 12 16

FIGURE 1.—TENSION CREEP CURVES.

not covered by the test results. This has led to a number of methods for interpreting the test data. For this purpose it is desirable to express a relationship for the creep deformation e_c in a time t for a unit stress s. When elevated temperatures must be considered, there is the additional variable of temperature.

Some of the methods for interpretation developed for elevated temperatures have been presented elsewhere.[2] More recently, procedures for interpretation of data have been proposed by S. H. Weaver,[4] C. R. Soderberg,[5] A. Nádai,[6] and others. A consideration of these methods shows that, although some of the methods fit a particular set of test data better than other methods, they do so by using a more complicated formula. If the creep-stress law is not a simple one, it leads to mathematical difficulties in developing a mechanics of creep, or the expressions deduced become too complicated for practical purposes. A creep law that satisfies the condition of simplicity and has considerable experimental support for many engineering materials at elevated temperatures is the one called the log-log creep-stress relationship. By this method, a constant creep rate is assumed. (See, for example, the many tests reported in "Compilation of Available High Temperature Creep Characteristics of Metals and Alloys," A. S. M. E.—A. S. T. M, 1938. These test results are plotted on a log-log basis and show an approximate linear plot.) In addition, a straight-line relationship is also assumed between the logarithm of the creep rate and the stress—that is,

$$C_n = a\, s^n \qquad (1a)$$

in which C_n = constant creep rate = $\frac{e_c}{t}$, and a and n are experimental constants. Equation 1*a* also can be written:

$$e_c = a\, s^n\, t \qquad (1b)$$

Considering the behavior of engineering materials at normal temperatures, there are only a few experiments upon which to base a creep-stress law. It is desirable to review briefly some of these tests.

[4] "The Creep Curve and Stability of Steels at Constant Stress and Temperature," by S. H. Weaver, Transactions, A. S. M. E., Vol. 58, 1936, p. 745.

[5] "The Interpretation of Creep Tests for Machine Design," by C. R. Soderberg, *ibid.*, p. 733.

[6] "The Influence of Time Upon Creep"—The Hyperbolic Sine Creep Law, Stephen Timoshenko 60th Anniversary Vol., by A. Nádai, Macmillan Co., 1938, pp. 155-171.

Experiments in tension [8] on Aluminum Alloy 3S-3/4H show that the log-log interpretation expressed by equations 1 is a good approximation. This is shown in Figure 2 where the logarithm of the constant creep rates obtained from Figure 1 are plotted versus the logarithm of the stress. The linear relation between these two quantities in Figure 2 can be expressed by either equation 1*a* or 1*b*.

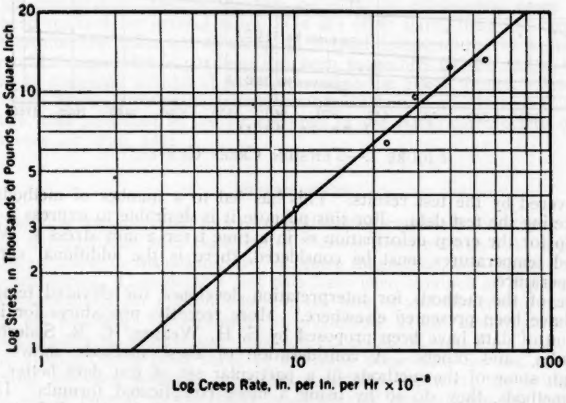


FIGURE 2.—RELATION BETWEEN CREEP RATE AND STRESS.

Creep tests on various engineering materials at normal temperatures were reported by R. G. Sturm, Assoc. M. Am. Soc. C. E., C. Dumont, and F. M. Howell.[7] These experiments were made on several aluminum alloys, soft and hard copper, and two steels. The material was in the form of sheets and wires. These tests show a log-log plot between the creep deformation and the time. Accordingly, the creep law proposed can be expressed essentially by the equation

$$e_c = F\, t^m \qquad (2)$$

in which F and m are experimental constants. By proposing equation 2, no attempt is made to give a general expression involving a variable stress.

Creep experiments on five steels were reported by H. J. French, H. C. Cross, and A. A. Peterson,[8] in which some of the tests were made at room temperature. These experiments were made on:

One 0.24 per cent carbon boiler-plate steel specimen;
One structural alloy steel specimen;
One tungsten-vanadium high-speed steel specimen;
One chromium "stain-resisting" steel specimen; and
One nickel chromium "austenitic" steel specimen.

Although sufficient experiments were not made for purposes of establishing a creep-stress law, these tests are significant in giving evidence of creep

[7] "A Method of Analyzing Creep Data," by R. G. Sturm, C. Dumont, and F. M. Howell, Transactions, A. S. M. E., Vol. 58, 1936, p. A-68.

[8] "Creep in Five Steels at Different Temperatures," by H. J. French, H. C. Cross, and A. A. Peterson, Paper No. 362, Technical Papers of the Bureau of Standards, Vol. 22, 1927-1928.

deformation in structural steel. They show that, for the structural steel tested, creep occurs when the stress is only slightly above the "proportional limit" of the material.

Creep of commercial leads at normal temperatures has been investigated by A. J. Phillips[9] and by H. F. Moore, B. B. Betty, and C. W. Dollins.[10] These experiments indicate that the log-log creep-stress law expressed by equations 1 is a good approximation. Although this material is not of general importance, it is of interest in showing that the log-log plot applies.

Concrete is an important structural material and therefore should also be considered, if possible, in formulating a creep-stress law. Many creep-stress experiments on concrete in compression have been made. A summary of some of the test results and interpretations is given by J. R. Shank,[11] M. Am. Soc. C. E. The recommended creep-stress law on the basis of a study by Mr. Shank,[11] in terms of equations 1 and 2, is

$$e_c = a\, s\, t^m \qquad (3)$$

An examination of the foregoing creep-stress equations shows that a general expression for these equations might be written

$$e_c = a\, s^n\, t^m \qquad (4)$$

Equation 4 seems to have considerable experimental support and has the advantage of simplicity when compared to some that have been proposed. One disadvantage of most creep-stress relationships, as in the case of equation 4, is that the elastic or initial deformation is not included. Referring to the deformation-time relation of Figure 3, the total unit deformation e at a time t is made up of the elastic or initial deformation e_o plus the creep deformation e_c—that is,

$$e = e_o + e_c \qquad (5a)$$

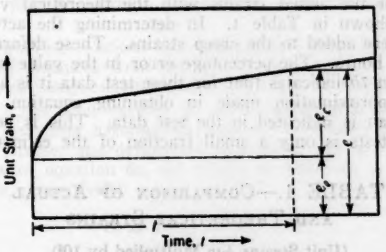


FIGURE 3.—CREEP-TIME CURVE.

The value of the elastic or initial deformation will be assumed equal to

$$e_o = B\, s^p \qquad (5b)$$

[9] "Some Creep Tests on Lead and Lead Alloys," by A. J. Phillips, Proceedings, A. S. T. M., Vol. 36, 1936, Pt. II, Technical Papers, p. 170.

[10] "The Creep and Fracture of Lead and Lead Alloys," by H. F. Moore, B. B. Betty, and C. W. Dollins, Bulletin No. 272, Univ. of Illinois Eng. Experiment Station, Urbana, Ill., 1935; see also "The Investigation of Creep and Fracture of Lead and Lead Alloys for Cable Sheathing," by the same authors, Bulletin No. 306, *ibid.*, 1938.

[11] "The Mechanics of Plastic Flow of Concrete," by J. R. Shank, Proceedings, A. C. I., 1936, pp. 149-180.

For a material in which Hooke's law applies, $B = 1/E$ and $p = 1$, in which E = the modulus of elasticity of the material. Substituting the values of the deformations from equations 4 and 5*b* in equation 5*a*, the unit deformation is

$$e = B\,s^p + a\,s^n\,t^m \quad \ldots\ldots (6a)$$

Or, for a material for which Hooke's law applies,

$$e = \frac{s}{E} + a\,s^n\,t^m \quad \ldots\ldots (6b)$$

Equations 6 define the total deformation in terms of the unit stress and the time. In addition, there are five experimental constants. This creep-stress law is used in the next section to develop a theory for stresses and deflections in the case of bending. It is found, however, that this leads to cumbersome expressions, making the formulation of structural mechanics in the case of creep very difficult. For this reason an approximation to equation 6*a* is suggested. This involves assuming the elastic or initial deformation to be expressed by $B\,s^n$ and determining B for the maximum stress—that is, B is determined from the equation $B\,s^2_{max} = s_{max}/E$. The total strain then will be assumed as

$$e = s^n\,(B + a\,t^m) \quad \ldots\ldots (7a)$$

An indication of the accuracy obtained in using equation 7*a* is given by an analysis of the test data shown in Figure 1. Using the results for specimens B3.2 and B3.1, $n = 1.22$, $m = 0.20$, $a = 3.75 \times 10^{-9}$, and $B = 1.04 \times 10^{-8}$. The value of B was determined on the basis of Hooke's law, as explained. The expression for the total strain as given by equation 7*a* then becomes

$$e = s^{1.22}\,(1.04 \times 10^{-8} + 3.75 \times 10^{-9}\,t^{0.20}) \quad \ldots\ldots (7b)$$

A comparison of the actual strains with the theoretical values given by equation 7*b* is shown in Table 1. In determining the actual strains, the elastic strains were added to the creep strains. These deformations are for a time $t = 1760$ hours. The percentage error in the value of the strains as given by equation 7*b* indicates that for these test data it is a good approximation. The approximation made in obtaining equation 7*a* produces a smaller error than is indicated in the test data. This is because the time covered by the tests is only a small fraction of the estimated life of the

TABLE 1.—COMPARISON OF ACTUAL AND THEORETICAL STRAINS

(Unit Strains Are Multiplied by 10^6)

Specimen	Unit stress (lb per sq in.)	Actual Strain			Theoretical strain (Eq. 7*b*)	% error
		Creep strain (e_c)	Elastic strain (e_e)	$e_c + e_e$		
B3.2	3,472	35	29	64	56	−12
D2.3	6,227	70	52	122	115	− 6
D2.1	9,655	90	81	171	163	− 5
D2.2	12,321	110	102	212	271	−28
B3.1	13,130	174	109	283	284	0

structure—that is, the test data cover a period of approximately two months, whereas the limiting deflection would be based on a time on the order of ten or twenty years. The elastic deformation is a very small fraction of the creep deformation, and the error produced in using B s^n for the elastic strain is therefore very small.

Members Subjected to Bending.

For a straight beam subjected to a pure bending moment M, as shown in Figure 4, a theory for the stress distribution can be developed as in the case

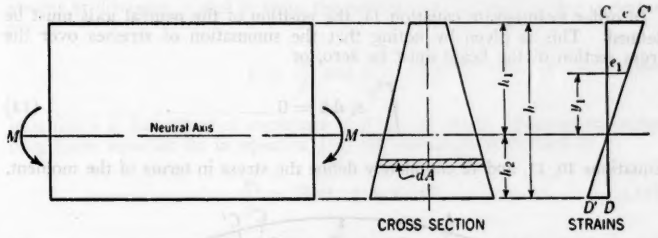


Figure 4.

of the elastic theory. To do this it is only necessary to replace Hooke's law in the development, using the elastic theory with the creep-stress law expressed by equation 6*a*—that is, the following assumptions will be made:

(1) A transverse cross section of the beam originally plane and normal to the center line of the member remains plane after bending;

(2) For any fiber the strains are defined as in the case of tension by equation 6*a*;

(3) The creep-stress relations are identical for fibers in tension and compression at equal distances from the neutral axis; and

(4) The lateral compression between the fibers is negligible.

Assumption (1) has been verified by some experiments in bending.[12] By means of the foregoing assumptions and the conditions of equilibrium, a theory for the stresses and deflections in bending can be obtained. By the creep-stress law of equation 6*a*, the strains at distances y_1 and h_1 from the neutral axis are, respectively,

$$e_1 = B\, s^p{}_1 + a\, s^n{}_1\, t^m \qquad (8a)$$

and

$$e = B\, s^p + a\, s^n\, t^m \qquad (8b)$$

Assuming that plane sections remain plane, the strains are proportional to the distances from the neutral axis and

$$\frac{e_1}{e} = \frac{y_1}{h_1} \qquad (9)$$

[12] "Stresses and Deformations in Pipe Flanges Subjected to Creep at High Temperatures," by J. Marin, Journal, Franklin Inst., Vol. 226, November, 1938, p. 645.

Placing values of the strains from equations 8 in equation 9,

$$y_1 = h_1 \left(\frac{B\, s^{p_1} + a\, s^{n_1}\, t^m}{B\, s^p + a\, s^n\, t^m} \right) \quad \ldots\ldots (10)$$

For equilibrium the external and resisting moments are equal, or

$$M = \int y_1\, s_1\, dA \quad \ldots\ldots (11)$$

In order to integrate equation 11, the position of the neutral axis must be defined. This is given by noting that the summation of stresses over the cross section of the beam must be zero, or

$$\int_{h_1}^{h_2} s_1\, dA = 0 \quad \ldots\ldots (12)$$

Equations 10, 11, and 12 completely define the stress in terms of the moment.

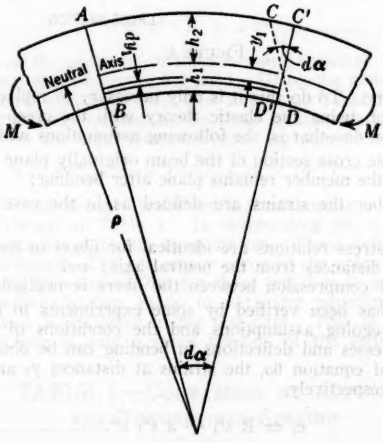


FIGURE 5.

For a rectangular cross section with width b and depth h, equation 12 shows that the neutral axis and centroid conincide. For a rectangular cross section $dA = b\, dy_1$, and equation 11 becomes

$$M = b \int y_1\, s_1\, dy_1 \quad \ldots\ldots (13)$$

To evaluate the integral in this equation it is necessary to determine dy_1 from equation 10. Placing the values of dy_1 and y_1 in equation 13 and integrating, the resulting equation is

$$M = \frac{b\,h^2}{2(B\,s^p + a\,s^n\,t^m)^2}$$

$$\times\left(\frac{p\,B^2\,s^{2p+1}}{2p+1} + \frac{2\,p\,n\,a\,B\,s^{p+n+1}\,t^m}{p+n+1} + \frac{n\,a^2\,s^{2n+1}\,t^{2m}}{2n+1}\right) \quad (14)$$

The stress s on the outer fiber is defined by equation 14 in terms of the moment M, dimensions of the beam, and constants of the material.

In order to obtain an expression for the deflection in a beam subjected to pure bending, consider a section of a beam as shown in Figure 5. The strains at distances y_1 and h_1 from the neutral axis in terms of the curvature are

$$e = \frac{h_1}{\rho} \text{ and } e_1 = \frac{y_1}{\rho} \quad (15a)$$

in which ρ is the radius of curvature of a curved beam. Placing the values of e_1 from equation 8*a* in equation 15*a*, the curvature is defined by

$$\frac{y_1}{\rho} = B\,s^p{}_1 + a\,s^n{}_1\,t^m \quad (15b)$$

Using the condition of equilibrium as given by equation 13, and placing the values of y_1 and dy_1 as obtained from equation 15*b* in this equation,

$$\frac{1}{\rho} = \frac{2}{h}\,(B\,s^p + a\,s^n\,t^m) \quad (15c)$$

For small deflections the relation between the deflection y and the curvature is

$$\frac{1}{\rho} = \frac{d^2y}{dx^2} \quad (15d)$$

The differential equation for the deflection in terms of the stress on the outer fiber is, by equations 15*c* and 15*d*,

$$\frac{d^2y}{dx^2} = \frac{2}{h}\,(B\,s^p + a\,s^n\,t^m) \quad (16)$$

A consideration of equations 14 and 16 for the stress and deflection in beams of rectangular cross section shows that these expressions are too complicated for practical purposes. If the value of p is assumed equal to n, the foregoing theory is simplified. This is equivalent to assuming the approximate creep-stress law given by equation 7*a*. Placing p = n in equation 14, the expression for the maximum stress is

$$s = \left(\frac{M\,h}{I}\right)\left(\frac{2n+1}{6n}\right) \quad (17)$$

in which $I = b\,h^3/12$ = the moment of inertia of the cross section.

The differential equation for the deflection is given by substituting the value of p = n in equation 16. Then

$$\left(\frac{D}{B + a\,t^m}\right)\frac{d^2y}{dx^2} = M^n \quad (18)$$

in which

$$D = \frac{(2b)^n \left(\frac{h}{2}\right)^{2n+1}}{\left(2 + \frac{1}{n}\right)^n} \quad \text{.....(19)}$$

For the case where $t = 0$ and $n = 1$, equation 18 reduces to the equation given by the elastic theory. The value of D can be obtained in a manner similar to the foregoing for symmetrical sections other than rectangular. Equation 18 can be used for determining deflections in beams in a manner similar to the usual double integration method. It is only necessary to integrate equation 18 twice and to obtain the constants of integration by using the boundary conditions of the particular problem. For example, in a simply supported beam of length l, subjected to a center concentrated load P, the deflection at a distance x from one end is

$$y = \frac{B + a\,t^m}{D}\left[\frac{x^{n+2}}{(n+1)(n+2)} - \frac{\left(\frac{l}{2}\right)^{n+1}}{(n+1)}\,x\right]\frac{P^n}{2^n} \quad \text{.....(20)}$$

For a value of $x = l/2$, the maximum deflection is, by inspection or by equation 20,

$$y_{max} = \frac{B + a\,t^m}{D} \times \frac{l^{n+2}}{2^{2(n+1)}} \times \frac{P^n}{n+2} \quad \text{.....(21)}$$

The accuracy of the foregoing theory is indicated by a comparison between the measured and theoretical deflections in aluminum alloy beams subjected to pure bending.[5] These beams were subjected to pure bending moments of values shown in Table 2. They were of rectangular cross section of width $b = 0.50$ inch and depth $h = 1.00$ inch, and the deflections were measured over a gage length $l = 8$ inches. By equation 18 the expression for the deflection becomes

$$y = \frac{B + a\,t^m}{D} \times \frac{l^2}{8} \times M^n \quad \text{.....(22)}$$

Using the values for the experimental constants obtained from the tension tests of Figure 1, the theoretical values of the deflections, as given by equation 22, are shown in Table 2. The experimental values of these

TABLE 2.—COMPARISON OF EXPERIMENTAL AND THEORETICAL DEFLECTIONS

Specimen No.	Moment (in-lb)	Theoretical deflection (in.)	Actual deflection (in.)	% error
D5	687	0.018	0.013	28
A3	996	0.029	0.022	24
C3	1,333	0.040	0.031	22
A4	1,667	0.053	0.046	13

deflections are also listed. A comparison between the theoretical and experimental values shows that the theory gives a good approximation. In making this comparison the preliminary nature of these tests in this new field of study must be considered.

Statically Indeterminate Structures.

Knowing the relationship between the stresses and deformations in the case of simple tension and pure bending, theories for the analysis of statically indeterminate structures can be developed. It is convenient to discuss first statically indeterminate beams of one span.

(1) *Double Integration Method—Beams of One Span.*—For beams that are statically indeterminate, the values of the reactions and deflections can be found in a manner similar to the double integration method used for the elastic case—that is, essentially it is only necessary to integrate the differential equation 18 twice, first substituting the value of the moment M in terms of x for the particular problem. The constants of integration and the statically indeterminate reactions are then found by using the particular boundary conditions of the problem. The difficulties in this procedure are mathematical because in some cases the integration is not direct and in others considerable work is involved. These difficulties arise since the moment M in equation 18 is raised to a power n that is usually different from one, and the simplifications afforded in the elastic case by using the methods of superposition can no longer be used in this analysis. Some examples of statically indeterminate beams will now be discussed and the maximum moments compared with the "elastic" values.

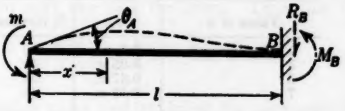


Figure 6.

Example 1.—A beam fixed at one end, simply supported at the other, and subjected to an end moment m is shown in Figure 6. By equation 18 the deflection equation is

$$C_n \frac{d^2y}{dx^2} = M^n = (R_A\, x - m)^n \qquad (23)$$

in which R_A = the reaction at point A, and

$$C_n = \left(\frac{D}{B + a\, t^m}\right) \qquad (24)$$

Integrating equation 23 twice,

$$C_n \frac{dy}{dx} = \frac{(R_A\, x - m)^{n+1}}{R_A\,(n+1)} + c_1 \qquad (25a)$$

and

$$C_n\, y = \frac{(R_A\, x - m)^{n+2}}{R^2{}_A\,(n+1)\,(n+2)} + c_1\, x + c_2 \qquad (25b)$$

in which c_1 and c_2 are constants of integration. Using the boundary conditions that the deflections are zero at both ends of the beam and the slope is zero at point B, the value of the reaction R_A is determined by the equation

$$R_A\, l\,(n+2)\,(R_A\, l - m)^{n+1} - (R_A\, l - m)^{n+2} + (-m)^{n+2} = 0 \quad \ldots\ldots (26a)$$

Noting that $R_A\, l - m = M_B$, equation 26*a* can be written in terms of M_B—that is,

$$M^{n+2}{}_B + \left(\frac{n+2}{n+1}\right) m\, M^{n+1}{}_B + \left(\frac{1}{n+1}\right)(-m)^{n+2} = 0 \quad \ldots\ldots (26b)$$

The value of the moment M_B as defined by equation 26*b* will depend upon the value of n. For different values of n the variation in M_B is represented in Table 3. The table also shows that there is a 44 per cent increase in the moment at B for n = 7 over the value determined by the elastic theory. Values of n at elevated temperatures for steel have been reported as high as 9 so that it is possible to have an appreciable increase in the value of the moments over those values obtained by elasticity. The value of the bending stress, as determined by equation 17 for a rectangular section, is also modified, depending upon the value of n. In estimating the resultant effect it is necessary, therefore, to consider the influence of the change in moments as well as the change in stress distribution. Another important consideration is that of limiting the maximum deflection obtained from equation 25*b*.

TABLE 3.—VALUES OF M_B

Value of n	M_B	% increase
1 (elastic case)	0.5 *m*	0
3	0.65 *m*	30
5	0.67 *m*	34
7	0.72 *m*	44

Example 2.—A uniformly loaded beam, fixed at one end and simply supported at the other, is shown in Figure 7. The differential equation for this problem, using equation 18, is

$$C_n \frac{d^2y}{dx^2} = \left(R_A x - \frac{w\, x^2}{2}\right)^n \quad \ldots\ldots (27)$$

in which w is the load per unit of length. Proceeding in a manner similar to Example 1, an equation for the reaction R_A at point A can be determined. For a value of n = 3 the equation for R_A is

$$R^3{}_A - (1.25\ w\ l)\ R^2{}_A + (0.535\ w^2\ l^2)\ R_A - 0.078\ w^3\ l^3 = 0 \quad \ldots\ldots (28)$$

From equation 28 the value of R_A is 0.40 w l. For the elastic case, $R_A = 0.375$ w l. The moment at the support for n = 3 is, by statics, $M_B = 0.10\ w\ l^2$ as compared to $0.125\ w\ l^2$ by the elastic case.

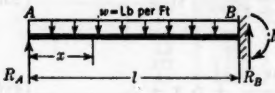


FIGURE 7.

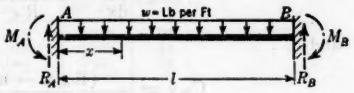


FIGURE 8.

Example 3.—For a beam fixed at both ends and subjected to a uniform load, as shown in Figure 8, equation 18 becomes

$$C_n \frac{d^2y}{dx^2} = \left(\frac{w\,l\,x}{2} - M_A - \frac{w\,x^2}{2} \right)^n \quad \ldots\ldots (29)$$

Again following the procedure as explained for Example 1—namely ,integrating equation 29 twice and applying the boundary conditions—an equation defining the value of M_A can be determined. For $n = 3$ this relation is

$$M^3{}_A - (0.25\ w\ l^2)\ M^2{}_A + (0.025\ w^2\ l^4)\ M_A - 0.00088\ w^3\ l^6 = 0 \ldots (30)$$

Solving for this moment, its value is $M_A = 0.074\ w\ l^2$ as compared with $M_A = 0.0833\ w\ l^2$ for the elastic case.

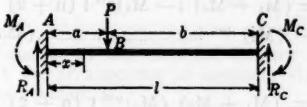


FIGURE 9.

FIGURE 10.

Example 4.—For a beam fixed at both ends and subjected to a concentrated load (Figure 9), it is necessary to write equations for both parts AB and BC of the beam. This leads to equations that are too complicated to present in this paper. If the load is considered at the center, $R_A = P/2$ and equation 18 need be written for only one half the beam. Thus, for AB,

$$C_n \frac{d^2y}{dx^2} = \left(\frac{P}{2} x - M_A \right)^n \quad \ldots\ldots (31)$$

Integrating this equation twice and applying the necessary boundary conditions, an equation for M_A can be found—namely,

$$\left(\frac{P\,l}{4} - M_A \right)^{n+1} - (-M_A)^{n+1} = 0 \quad \ldots\ldots (32)$$

Using this relation, $M_A = 0.13\ P\ l$ for $n = 3$, and $M_A = 0.14\ P\ l$ for $n = 5$. These values are higher than that of $0.125\ P\ l$ as found for the elastic case.

The foregoing examples show that the values of the statically indeterminate reactions and moments may be appreciably different from the magnitudes obtained for the elastic case. It may be necessary, therefore, to consider this difference in the design of statically indeterminate beams in which creep occurs.

(2) *Slope-Deflection Method*—The slope-deflection method is one of the classical methods for structural analysis. For the condition of creep, equations similar to those for the elastic case can be derived. To obtain these equations, consider the beam AB in Figure 10 simply supported at points A and B and subjected to moments M_1 and M_2 at the ends. By the slope-deflection method these moments are determined in terms of the slopes or rotations θ_1 and θ_2 at the ends of the beam. To do this, equation 18 can be used. Then

$$C_n \frac{d^2y}{dx^2} = M^n = (R_1\,x - M_1)^n \quad \ldots\ldots (33)$$

Integrating this equation twice,

$$C_n \frac{dy}{dx} = \frac{(R_1 x - M_1)^{n+1}}{R_1 (n+1)} + c_1 \quad \ldots\ldots (34a)$$

and

$$C_n y = \frac{(R_1 x - M_1)^{n+2}}{R_1^2 (n+1)(n+2)} + c_1 x + c_2 \quad \ldots\ldots (34b)$$

Using the conditions that the deflections are zero at both $x = 0$ and $x = l$, the constants of integration c_1 and c_2 are determined. Then the slopes ϑ_1 and ϑ_2 are the values of dy/dx at $x = 0$ and $x = l$, respectively, as found by equation 34*a*. Noting also that $R_1 = (M_1 + M_2)/l$, the slope-deflection equations expressing the slopes in terms of the moments are:

$$C_n \vartheta_1 (M_1 + M_2)^2 (n+2)(n+1) = (M_1 + M_2)(-M_1)^{n+1} l (n+2) + l (-M_1)^{n+2} - (M_2)^{n+2} l \quad \ldots\ldots (35a)$$

and

$$C_n \vartheta_2 (M_1 + M_2)^2 (n+2)(n+1) = (M_1 + M_2)(M_2)^{n+1} l (n+2) + l (-M_1)^{n+2} - (M_2)^{n+2} l \quad \ldots\ldots (35b)$$

The solution of equations 35 simultaneously for M_1 and M_2 offers difficulties not present in the elastic case. The conclusion must be, therefore, that the slope-deflection method is not suitable and that the modified moment-distribution method should be used.

(3) *Moment-Distribution Method.*—A study of the moment-distribution method of structural analysis as applied to members subjected to creep shows again the fundamental nature of the Hardy Cross method.[18] The basic character of this classic method remains unchanged for applications where creep must be considered, but it is necessary to determine the beam constants—namely, the fixed-end moments, the carry-over factors, and the distribution factors.

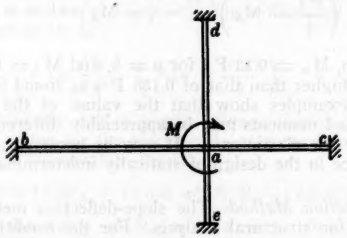


FIGURE 11.

The determination of the fixed-end moments has been explained herein in the section on the double integration method. To determine the carry-over and distribution factors it is necessary to determine the slope at the left end of the beam in Example 1, as shown in Figure 6. This slope is the value of dy/dx in equation 23 for the value $x = 0$ or

[18] Transactions, Am. Soc. C. E., Vol. 96 (1932), p. 1.

$$C_n\vartheta_A = m^n\left\{\frac{1\,[(-1)^{n+1} - (k)^{n+1}]}{(k+1)\,(n+1)}\right\} \quad \text{.....(36)}$$

or

$$m^n = \frac{C_n\,(k+1)\,(n+1)}{1\,[(-1)^{n+1} - (k)^{n+1}]}\,\vartheta_A = J\,\vartheta_A \quad \text{.....(37)}$$

in which

$$k = \frac{M_B}{m} \quad \text{.....(38}a\text{)}$$

and

$$J = \frac{C_n\,(k+1)\,(n+1)}{1\,[(-1)^{n+1} - (k)^{n+1}]} \quad \text{.....(38}b\text{)}$$

Equation 37 expresses the moment m at A in terms of the slope ϑ_A at A. If the arrangement of members shown in Figure 11 is considered, then, by equation 37,

$$M^n_{ab} = J_{ab}\vartheta_a;\ M^n_{ac} = J_{ac}\vartheta_a;\ M^n_{ad} = J_{ad}\vartheta_a;\ \text{and } M^n_{ae} = J_{ae}\vartheta_a \quad \text{.....(39)}$$

For equilibrium,

$$M_{ab} + M_{ac} + M_{ad} + M_{ae} + M = 0 \quad \text{.....(40)}$$

Substituting values from equation 39 in equation 40, and solving for ϑ_a,

$$\vartheta_a = -\frac{M^n}{(\Sigma J^{1/n})^n} \quad \text{.....(41)}$$

Placing this value of the slope ϑ_a in equation 39, the ratios of the moments in the members to the applied moments—called the distribution factors—are:

$$\frac{M_{ab}}{M} = \frac{J^{1/n}_{ab}}{\Sigma J^{1/n}};\ \frac{M_{ac}}{M} = \frac{J^{1/n}_{ac}}{\Sigma J^{1/n}};\ \frac{M_{ad}}{M} = \frac{J^{1/n}_{ad}}{\Sigma J^{1/n}};\ \text{and } \frac{M_{ae}}{M} = \frac{J^{1/n}_{ae}}{\Sigma J^{1/n}} \quad \text{.....(42)}$$

or

$$D_f = \frac{J^{1/n}_p}{\Sigma J^{1/n}} \quad \text{.....(43)}$$

in which D_f is the distribution factor for any member.

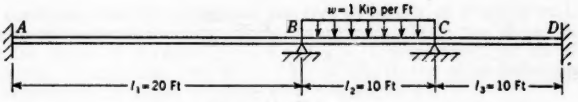
The carry-over factor can be obtained by reference to Figure 10 and equation 35*b*, which give the slope at point B in the beam AB. If this slope, $\vartheta_2 = \vartheta_B$, is placed equal to zero, the moment M_2 is then the moment at the fixed end of a beam AB, fixed at point B, simply supported at A, and subjected to a moment M_1 at A. The ratio M_2/M_1 is then the carry-over factor.

By equations 35*b*, with $\vartheta_2 = 0$, the carry-over factor, $C_f = M_a/M_1$, is defined by the equation:

$$C^{n+2}_f + \left(\frac{n+2}{n+1}\right) C^{n+1}_f - \left(\frac{1}{n+1}\right) = 0 \quad \text{.....(44)}$$

For values of n from 1 to 7 the carry-over factor varies from 0.5 to 0.72. With the fixed-end moments, distribution factors, and carry-over factors determined, the solution of some particular problem can be found, as illustrated by the following example.

Example 5.—A continuous beam of three spans is shown in Figure 12. The moments at the supports will be determined by the moment-distribution method for a value of n = 3, and these values will be compared with those for the elastic case.



(a) Computation for Moments at Supports

C_f	0.61		0.61		0.61	
D_f		0.44	0.56	0.50	0.50	
Fixed-End Moments			−7.40	−7.40		
		−3.26	+4.14	+3.70	−3.70	
	+1.98		−2.26	−2.53		+2.26
		−1.00	+1.26	+1.26	−1.26	
	+0.61		−0.77	−0.77		+0.77
		−0.34	+0.43	+0.38	−0.38	
	+0.21		−0.23	−0.26		+0.23
		−1.0	+0.13	+0.13	−0.13	
Moments at Supports	+2.80	−4.70	−4.70	−5.47	−5.47	+3.26

(b) Comparative Computations for the Elastic Case

C_f	0.5		0.5		0.5	
D_f		0.33	0.67	0.50	0.50	
Fixed-End Moments			−8.83	−8.83		
		−2.78	+5.55	+4.16	−4.16	
	+1.39		−2.08	−2.78		+2.08
		−0.68	+1.40	+1.39	−1.39	
	+0.34		−0.70	−0.70		+0.70
		−0.23	+0.47	+0.35	−0.35	
	+0.12		−0.18	−0.24		+0.18
		−0.06	+0.12	+0.12	−0.12	
Moments at Supports	+1.85	−3.75	−3.75	−6.02	−6.02	+2.96

FIGURE 12.—CONTINUOUS BEAM OF THREE SPANS

In order to determine the moments at the supports, the beam constants must be obtained. The fixed-end moments for a uniform load and a value of n = 3 are determined in Example 3. These moments are $M = 0.074\ w\ l^2 = 0.074 \times 1 \times 100 = 7.40$ kip-ft. The carry-over factor for a value of

$n = 3$ is determined by equation 44. Its magnitude is $C_t = 0.61$. Finally, the distribution factors are found by equation 43, using equation 38*b* to determine J_p, and equation 38*a* to find k. In Example 1 the value of $k = 0.65$ is given for a value of $n = 3$. By equation 38*b*, $J_{ab} = 8.02\ (C_n/l) = (8.02/l_{ab})\ C_n$. Similarly, $J_{bc} = 8.02\ (C_n/l_{bc})$.

The distribution factor for member AB is now calculated by equation 43—namely,

$$D_{ab} = \frac{(J_p)^{1/n}}{\Sigma(J^{1/n})} = \frac{J^{1/n}_{ab}}{J^{1/n}_{ab} + J^{1/n}_{bc}} = \frac{(1/l_{ab})^{1/n}}{(1/l_{ab})^{1/n} + (1/l_{bc})^{1/n}} \quad \ldots\ldots\ldots (45)$$

or $D_{ab} = 0.44$; and similarly, $D_{bc} = 0.56$. Using the beam constants as given in Figure 12(*a*), the moments at the supports are found. In Figure 12(*b*) these moments are determined for the elastic case for the purpose of comparison. A comparison of these moments indicates that there is sufficient difference in the moments to warrant consideration in design. Other types of statically indeterminate structures can be analyzed in a similar manner.

A consideration of the energy methods of structural analysis shows that these methods cannot be applied in the case of creep because they require that the material be elastic. Other fundamental methods of structural analysis used in the elastic theory do not apply in the case of creep as, for example, the use of the law of superposition and the determination of influence lines by mechanical means. On the other hand, secondary stresses in the case of creep can be determined since the moment-distribution method can be applied.

Conclusion.

In this paper a new creep-stress law is proposed for materials in which creep at both normal and elevated temperatures is important. Some experimental evidence is given and other data are cited that show the validity of the creep law assumed. A theory for stresses and deflections in members subjected to bending is developed and the values compared with creep test results. Theories are then derived for the analysis of statically indeterminate structures in cases where creep is present. Comparison of results with the elastic case shows that there may be an appreciable difference in the values of reactions and moments. It should be noted that there is considerable need for experimental work on this problem. In the meantime, however, this paper shows, in general, how structural analysis is modified when creep occurs in engineering materials.

APPENDIX.

Notation.

The following letter symbols, used in this paper, conform essentially to American Standard Letter Symbols for Mechanics, Structural Engineering and Testing Materials, prepared by a Committee of the American Standards Association, with Society representation, and approved by the Association in 1932:[14]

[14] ASA—Z10a—1932.

A = area; dA = element of area;
a = experimental constant; also B, F, m, n, and p;
B = (see a)
b = breadth;
C_n = constant creep rate: C_r = carry-over factor;
c_1 and c_2 = constants of integration;
D = a dimension constant: D_r = distribution factor;
E = modulus of elasticity;
e = strain, unit deformation;
F = (see a);
h = height or depth of beam:
h_1 = distance from neutral axis to top of beam;
h_2 = distance from neutral axis to bottom of beam;
I = moment of inertia;
J = (see equation 38*b*): J_p, J_{ab}, etc., refer to given members of a structural arrangement;
k = a substitution constant (equation 38*a*);
l = span length: l_1, l_2, etc., denote lengths of adjacent spans 1, 2, etc:
M = bending moment, subscripts denoting points about which moments are taken;
m = (see a);
n = (see a);
P = concentrated load;
p = (see a);
R = reaction at the point denoted by subscript;
s = unit stress;
t = time;
w = uniformly distributed load;
x = coordinate distance from the left support;
y = ordinate to an elastic curve; deflection of a beam;
α = angle of curvature between two points on a curved beam;
θ = slope of the elastic curve at points designated by subscripts; and
ρ = radius of curvature of beam.

EFFECT OF NOZZLE AND BUCKET DEPOSITS ON TURBINE CAPACITY AND EFFICIENCY.

This article by Mr. B. O. Buckland is reprinted from the February, 1942, issue of the General Electric Review. Mr. Buckland is connected with the Turbine Engineering Department of the General Electric Company. This material was presented as a paper at the November 4, 1941, meeting of the Engineers Society of Western Pennsylvania.

When salt deposits accumulate on the nozzles and buckets of a turbine; the steam-flow areas are reduced, the nozzle and bucket surfaces are roughened, and the distribution of energy in the turbine stages is disturbed by the presence of the deposit. These conditions result in a reduction in turbine capacity and efficiency.

The extent to which one type of deposit may accumulate is illustrated in Figures 1 and 2. This type is water soluble, but another that is not soluble may likewise accumulate to such an extent as to decrease both the capacity and efficiency of the turbine. A third type, also insoluble and illus-

trated in Figure 3, has little effect on the performance of the turbine because it has a hard smooth surface and rarely forms in more than film thickness.

Most of the information presented in this article is based on the writer's observations made in a number of turbine plants while making acceptance tests and turbine inspections during overhaul periods. The subject is here divided into five topics, as follows:

(1) Prevalence of the deposits.
(2) Appearance and location in the turbine.
(3) Resultant reduction in efficiency and capacity.
(4) Means of removing deposits.
(5) Causes and prevention.

Prevalence of the Deposits.

A great deal of work has been done on the problem presented by deposits and carryover, both by chemists and by boiler manufacturers. The chemists have contributed by providing methods of boiler-water treatment to reduce carryover and to minimize the deposits that may result from a given amount of carryover.[1 2 3] The boiler manufacturers have provided steam washers and improved water separators within their boilers to reduce the amount of the carryover.[4 5]

The results of the research chemists' work in the hands of persistent and careful plant operators can accomplish a great reduction in turbine deposits, as is shown by the experience of a plant where initially the amount and rate of accumulation of deposits were extremely aggravating. Figures 1 and 2 show nozzles and buckets of the high-pressure turbine of a compound condensing unit in this plant as they appeared when the turbine was opened for inspection in 1931. The smooth patches show where the deposit was broken off for the purpose of measuring its approximate thickness. Tests of the complete unit before the machine was opened—and with the condition of the high-pressure turbine as shown in Figures 1 and 2—gave results 20 per cent low in capacity and 3 to 6 per cent high in heat rate. The best that could be done at this time to relieve the situation was to wash the turbine every four or five weeks. A brief outline of this plant's experience with deposits is given near the end of this article. It is sufficient to say at this point that in 1939 (8 years later) the writer again saw this turbine opened for inspection, and at about the same time he saw the newer and higher pressure (1200-psi) turbine also opened. In the older turbine the blading had only a film of deposit, almost thin enough to be transparent. The newer turbine had only a slight amount of deposit in the high-pressure section. The plant operators had been able to produce these results and discontinue their washing practice by the application of the results of Straub's research.[1]

As the foregoing example indicates, progress in the control of deposits has been made. It is not implied, however, that such satisfactory control has been found to exist throughout the power-generation industry, at least not within the limits of the writer's experiences in testing and inspecting

[1] "Cause and Prevention of Steam Turbine Blade Deposits," by Straub, ASME Trans., vol. 57, p. 447, Nov. 1935; also Univ. of Ill. Bull., no 282, May 1936.

[2] "Solid Matter in Boiler Water," by Faulk and Ulmer, Ind. and Eng. Chem., vol. 30, p. 158, 1938.

[3] "Foaming of Boiler Water," by Faulk and Ryznar, Ind. and Eng. Chem., vol. 31, p. 722, 1939.

[4] "Mechanical Purification of Steam within the Boiler Drum," by Baker, ASME Trans., vol. 61, p. 711.

[5] "Seam Contamination: II," by Powell, Combustion, Oct. 1937, p. 27.

turbines. A review of the reports which the writer and his associates have made on turbine tests and inspections since 1935 reveals the data shown graphically in Figure 4 and itemized in the following:

(1) Thirty turbines were tested and 21 inspected, and of these 18 were both inspected and tested.

(2) Twelve of the 21 turbines inspected showed blade deposits on inspection and 9 were clean.

(3) Two of the 9 turbines inspected and found clean on inspection were found to accumulate deposits during the tests following the inspection. The appearance of the blading during inspection and other evidence in the case of these two showed that the turbine had been washed in the process of shutting down before the inspection.

(4) Seven of the 21 turbines inspected showed evidence of accumulating blade deposit during the period of the test, after they had been cleaned and reassembled.

(5) Twelve turbines were tested without being inspected. Three showed evidence that blade deposit existed at the time of the test.

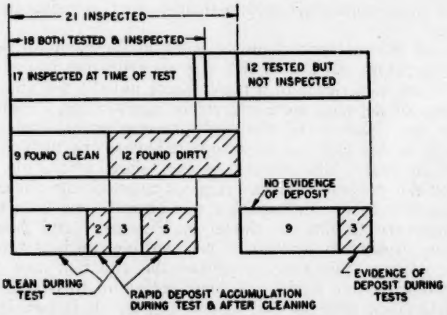


FIGURE 4.—PREVALENCE OF NOZZLE AND BUCKET DEPOSIT IN TURBINES AS FOUND BY TESTS AND INSPECTIONS.

For the purpose of the foregoing classification, a turbine was considered to have deposit if four or more stages showed deposit of 10 mils thickness or greater. The turbines considered dirty ranged from this minimum to a maximum deposit similar to that shown in Figures 1 and 2.

From the figures given, it will be noted that half of the turbines on which acceptance tests were made show evidence of deposit.

A dirty condition presents difficulties not only for the user but also for the builder. The guarantees are made with the understanding that the turbine will be tested in a clean condition, and no accurate procedure is available for correcting the results of a test for the effect of dirt. In turbines which are considerably low in capacity and high in shell pressure, it is evident that the turbine is either dirty or damaged and should either be washed or opened for cleaning and inspection before an acceptance test.

In cases where the amount of the deposit is relatively small, such as to produce not more than 2 or 3 per cent increase in shell pressure, it is difficult to establsh whether there is any deposit. Very often the most satisfactory procedure is to make the acceptance test after the turbine has

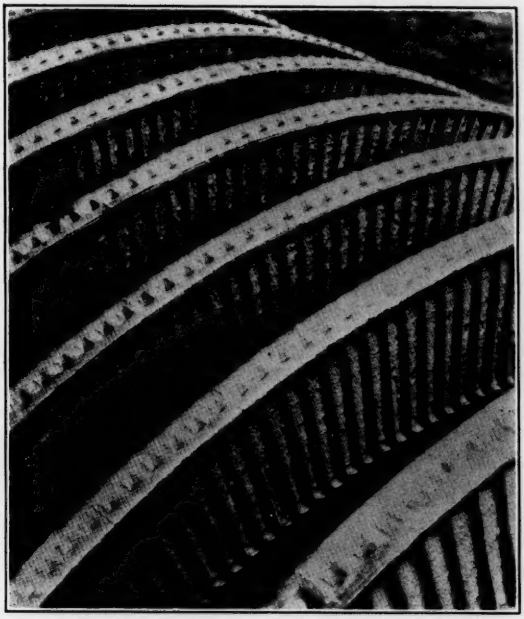


FIGURE 1.—EXAMPLE OF THE EXTENT TO WHICH A DEPOSIT MAY FORM ON THE BUCKETS OF A TURBINE.

Analysis:

Sodium silicate	27.70	Sodium chloride	1.58
Ferrous sulphate	1.58	Magnesium chloride	0.57
Sodium sulphate	3.17	Sodium hydroxide	18.11
Sodium phosphate	1.20	Sodium carbonate	12.68

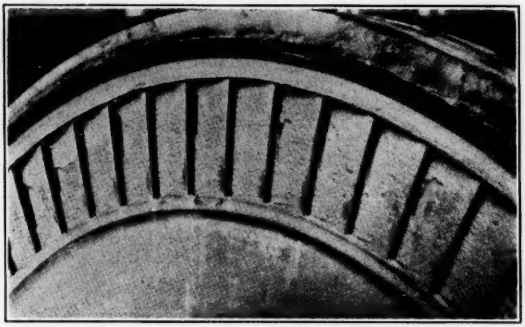


FIGURE 2.—DEPOSIT ON THE NOZZLE PARTITIONS OF THE TURBINE OF WHICH THE BUCKETS ARE SHOWN IN FIGURE 1.

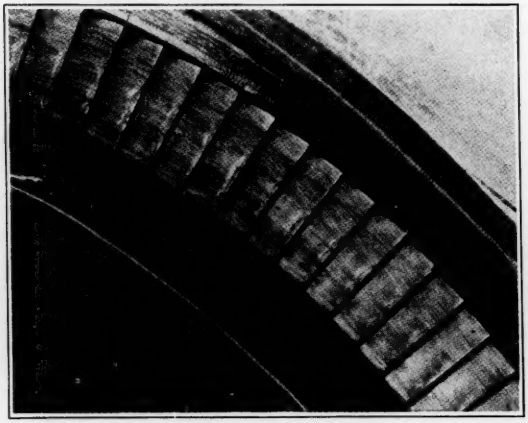


FIGURE 3.—A THIN HARD FILM TYPE OF DEPOSIT ON NOZZLE PARTITIONS.

been opened for its first inspection. Even when this is done it is often found that the deposits accumulate in the turbine so fast that test points run at the usual rate require too long a time to allow a consistent performance curve to be obtained. No satisfactory method of determining the performance exists in these cases. If the test must be made before the deposit problem is solved, the only approach to the situation seems to be to wash the turbine and to run test points as rapidly as possible immediately after washing.

Appearance and Location in the Turbines.

Most blade deposits can be classified, according to their general appearance, into three kinds:

(1) Friable and somewhat soft deposit.
(2) Sparkling silvery deposit.
(3) Smooth hard film.

The first of these ranges in color from pinkish white to a tan and is sometimes deliquescent. It is soft enough to be easily crushed and flaked off the blades in chunks. It occurs usually in greater quantities than the other two and can be readily removed by a jet of air and water. Usually. it occurs in the turbine from and including the first stage to the dew point in the turbine. This type of deposit is illustrated in Figures 1 and 2.

The second kind is silvery gray and sparkles in the light like new snow. It is hard and granular and clings tenaciously to the blading. Soaking in water or subjecting it to jets of air and water have little or no effect. It occurs from the third or fourth stage of a 900 F. turbine to a point in the turbine at which the temperature is 400 to 500 F. The proportion of this deposit which is formed on the surfaces other than the nozzles and buckets, such as wheel sides, diaphragm sides, etc., is greater than in the case of the other two kinds of deposit. Often sufficient deposit occurs on the diaphragm sides to give them the same sparkling appearance as the blading.

The third kind of deposit is a hard, relatively smooth, red or dark brown film streaked with white. It is very thin, though usually thick enough to be opaque. Very distinct traces of flow lines in the boundary layers of the nozzle and bucket passage are left in this deposit. These are illustrated in Figure 3.

Both the first and second kinds of deposit are found to some extent in the wakes where the stream spills over the leading edges of the nozzles and buckets and infrequently on the concave surfaces, but they seem to prefer to occur in far greater thickness at the points of low pressure in the turbine passages. The thickness on the convex surfaces—that is, on the inside of the fluid turn—is far greater than at other points in the passages. This cannot be due to jet separation on the convex surfaces as suggested by Soderberg[6] because the deposit occurs in regions where the pressure is dropping in the direction of flow and also because such jet separation as would have to occur to explain the deposit would be entirely too large and general in extent to be reasonable. At the points where jet separation would be expected to occur (in regions of recompression), the deposit is usually thinner than at points of rapid expansion or it is absent altogether.

[6] "Scaling of Turbine Blading," by Soderberg, Power, vol. 80, p. 596, Nov. 1936.

Reduction in Efficiency and Capacity.

The accumulation of deposit in a turbine has three effects which are measurable when the turbine is assembled and in operation:

(1) Increase in shell pressure.

(2) Reduction in efficiency.

(3) Reduction in capacity.

It would be convenient if the reduction in efficiency could be related to one or both of the other two effects. The reduction in capacity is the result of a combination of the reduction in flow and efficiency. The reduction in flow is very much dependent on where the deposits occur. Deposits in the first and early stages of a turbine reduce the flow to a greater amount than deposits in the later stages, but the effect on efficiency is probably independent of the location of the deposit for a particular amount and extent of deposit.

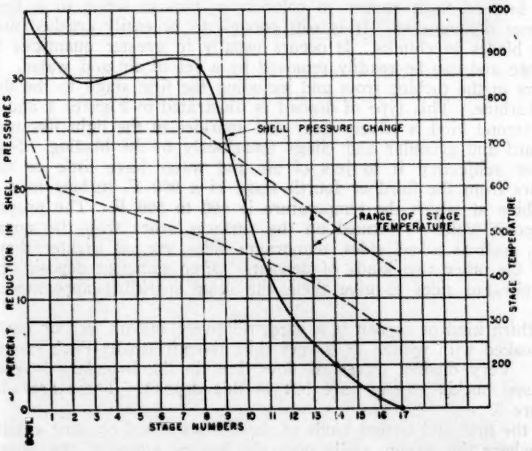


Figure 5.—Reduction in Shell Pressure Due to Removal of Blade Deposit from High-Pressure Turbine. Test Made with 665 psi abs. 730 F. at Turbine Throttle.

The effect of deposit on the efficiency has been measured in a number of instances. In one case a test was made with the turbine dirty, after which it was opened and cleaned and retested. The deposits accumulated slowly enough on the second test to permit a fairly consistent set of data to be obtained. The increase in shell pressures produced by the deposit is shown in Figure 5 and the change in efficiency in Figure 6. If the per cent increase in shell pressure is averaged by weighing it with the energy at the stage in which it occurs, the change in efficiency in Figure 6 will amount to about 0.6 per cent in efficiency for each 1 per cent change in this weighted average shell pressure.

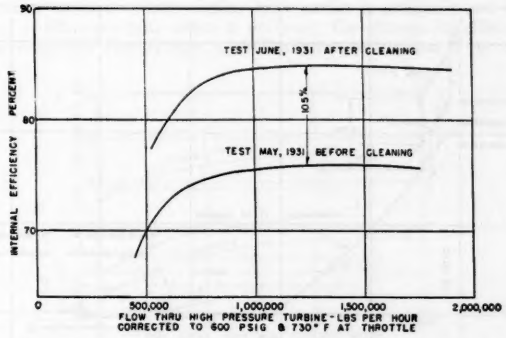


FIGURE 6.—IMPROVEMENT IN EFFICIENCY OF HIGH-PRESSURE TURBINE DUE TO REMOVAL OF DEPOSIT ON NOZZLES AND BUCKETS.

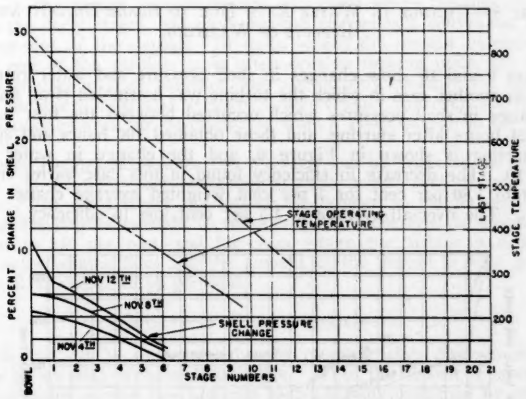


FIGURE 7.—DIFFERENCE IN SHELL PRESSURES BEFORE AND AFTER WASHING.

In a case in which the turbine was assembled clean, the deposits accumulated to such an extent during the period of test that the first-stage shell pressure was increased 7 per cent and the efficiency was reduced 1¼ per cent at high loads and 4.7 per cent at low loads. The turbine was then washed and several consistent tests were obtained quickly and immediately after washing. The shell pressures and water rates before and after washing are shown in Figures 7 and 8. On the foregoing basis of weighting the per cent increase in shell pressure, the change in water rates in Figure 8 varies from 0.6 per cent at high loads to 0.9 per cent at low loads for 1 per cent average change in shell pressures.

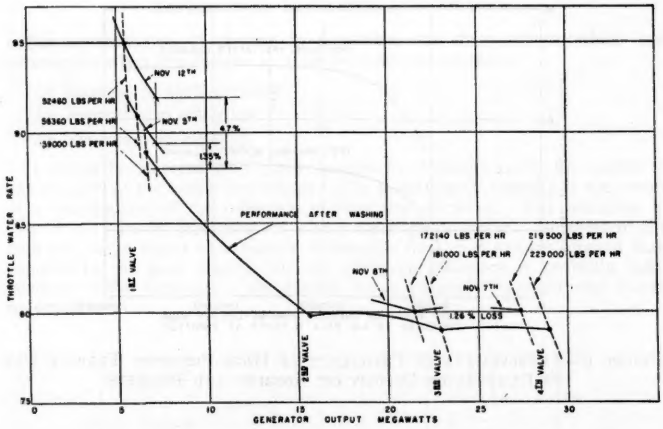


FIGURE 8.—CHANGE IN WATER RATE DUE TO BLADE DEPOSIT AND THE EFFECTS OF WASHING.

Figures 9 and 10 show changes in shell pressure and water rate due to deposit in another case in which the turbine was assembled clean and tested. The change in shell pressures which occurred between the first test point, within 24 hours after starting, and those obtained 160 hours and 200 hours after starting is shown in Figure 9, and the change in water rate in Figure 10. The decrease in efficiency found in this case varies from 1.22 per cent to 1.80 per cent for 1 per cent weighted average change in shell pressure. The over-all average is 1.5 per cent loss in efficiency.

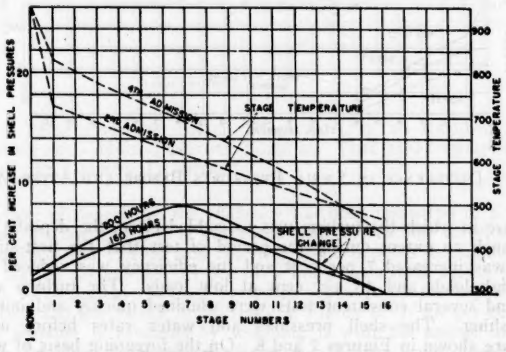


FIGURE 9.—CHANGE IN SHELL PRESSURE DUE TO ACCUMULATION OF BLADE DEPOSIT. THROTTLE STEAM CONDITIONS, 2015 PSI ABS. 950 F.

In two other cases in which similar information is available and in which the change in efficiency was about 2 per cent, the change in efficiency per weighted average per cent change in shell pressure was about 1 per cent.

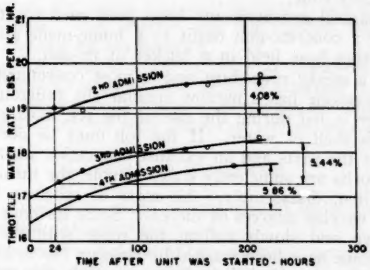


FIGURE 10.—INCREASE IN WATER RATE WITH TIME DUE TO TURBINE-BLADE DEPOSIT.

The weighted average change in shell pressure is not always a good measure of the change in performance. In cases where the deposit is confined to the early stages, the shell-pressure change is a convenient measure of the amount of the deposit. In cases where the deposit extends to the exhaust end of the turbine, as it often does in topping units, the average per cent change in shell pressure does not directly measure the amount of the dirt because the exhaust pressure is held fixed. However, the change in effective passage area can be calculated from a given change in the pressure distribution.

For very rough calculations, the efficiency can be regarded as decreasing about 1 per cent for each weighted average 1 per cent change in shell pressure. For cases where the turbine is designed to handle small volume flow and the deposit extends to the exhaust, the decrease in efficiency will be closer to 1½ per cent, and for cases where the turbine is designed to handle large volume flow and the amount of the deposit is very great, the figure will be closer to ½ per cent.

MEANS OF REMOVING DEPOSITS.

The deposit can be removed either by cleaning after disassembly of the parts or by washing without disassembly. When the deposit is more or less soluble, cleaning during overhaul can be readily accomplished by subjecting the nozzles and buckets to a strong jet of water and compressed air. When the deposits are insoluble and tenacious, as are silica deposits, the most satisfactory method of cleaning is to blast the blades with a jet of air into which fly-ash is fed in amounts similar to the amount of sand used in stand-blasting.[7]

The ash need not be as fine as that which floats out of the stack with the flue gases. The ash found in the base of the stack is satisfactory, although it may be in the form of shells and bubbles of fairly large size. Even river silt has been used with satisfactory results. Normal sand cuts too fast and leaves scratches, but fly-ash removes the deposit and leaves a smooth satiny

[7] "Fly-ash for Cleaning Turbine Blades," by Luxemberg, Power Plant Eng., vol. 38, p. 408, Sept. 1934.

finish on the blading. In some cases when the ash contains unburned carbon, a high graphite-like polish is obtained. It may be possible that this type of polish is beneficial in preventing deposits from sticking to the blades, at least for a short time.

Various mechanical arrangements have been used for fly-ashing, ranging from the use of a concrete-gun outfit to a home-made air-jet injector with the injector suction hose held in a bucket of fly-ash.

Fly-ashing is a messy procedure and is most conveniently done outdoors. However, if the nozzle has a narrow annular slit (surrounding the air jet) into which water is fed during the use of the jet, most of the dust can be trapped with this shell of water. If the job must be done indoors, a hood can be built over the parts and an exhaust connection made to outdoors.

When the deposits are sufficiently water soluble, the turbine may be cleaned by washing without disassembly. All sorts of schemes for washing have been tried, with varying degrees of success. Some turbines have been washed by shutting down and slowly rolling the rotor with the lower half shell full of water. Some have been washed by running the turbine at no load and shutting off the fire in the boiler, thus operating the turbine for a short period with saturated steam. And in a number of cases, merely shutting down the turbine over a week end has resulted in an increase of capacity, presumably because during the stopping and starting operations the turbine nozzles and buckets are subjected to moist steam or water vapor.

However, the most satisfactory method of washing, in the writer's opinion, is to set up the following apparatus and procedure: A nozzle of the proper size to pass sufficient steam flow at normal boiler pressure to run the turbine at a little more than one-fourth speed is provided in a line which bypasses the boiler stop valve and empties into the turbine inlet pipe ahead of the throttle and strainer. There is also provided a second nozzle designed to pass a flow somewhat more than sufficient to completely desuperheat the steam delivered by the first nozzle. The second nozzle is arranged to discharge into the bypass line at the point of exit of the first nozzle. A recording temperature-indicating instrument is provided with its bulb or thermocouple located in the turbine inlet pipe as far as possible from the point at which the bypass nozzle empties into this pipe but ahead of the throttle and strainer. The bulb should preferably be located on the underside of the pipe.

With this apparatus or its equivalent, the washing procedure is as follows: The turbine is unloaded and reduced in speed to about one-fourth or one-fifth normal, after which the flow required to maintain this speed is transferred to the bypass washing nozzle by shutting the boiler stop valve and opening the bypass nozzle line. The control of turbine speed is then transferred from the turbine throttle or control valves to the valve in the bypass line by opening the turbine throttle and control valves wide and adjusting the flow to hold the desired speed. This assists in carrying the moisture by keeping the velocity high and prevents superheating due to throttling.

Feed water is then admitted to the bypass line in such an amount as to reduce the steam temperature to the turbine at a rate not faster than 100 F. per hour until the saturation point is reached. The pressures and pressure drops in the washing nozzles are noted, and the feed-water flow is increased sufficiently to produce approximately 1 or 2 per cent moisture in the steam. The turbine is run under these conditions until it is judged clean on the basis of previous experience or from the conductivity of the condensate. After washing, the procedure is reversed until the turbine is on the line, care being taken not to exceed a reasonable rate of temperature rise, such as 100 F. per hour.

Literature on the subject records two instances of the washing of turbines with fluids other than steam and water. In one case two of three badly plugged 350-HP. turbines were washed with an inhibited hydrochloric acid solution.[8]

The composition of the scale was approximately one-fourth silica and three-fourths carbonate. The process was not entirely successful and the third turbine was cleaned by sandblasting.

In the other case the washing of turbines at standstill with 10 to 12 per cent sodium hydroxide solution and steam is reported to have been tried in Germany.[9] This was for the purpose of removing insoluble silica deposits. It is not clear from the report what degree of success was attained.

Causes and Prevention.

A theory of the causes of blade deposit should include explanations for the following observed characteristics:

(1) That soluble sodium hydroxide type deposits occur in some turbines and insoluble silica type deposits occur in others.

(2) That the deposit is found to begin in the first stage in some turbines and does not appear until the third or fourth stage in others.

(3) That heavier deposit occurs in the nozzle throats and other regions of low pressure.

The theory that sodium hydroxide in the steam is the chief cause of accumulation of deposits in turbines has been proposed by Straub.[1] He points out that sodium hydroxide can be carried in the superheated steam to the turbine in the form of concentrated solutions containing 10 to 20 per cent water. These solutions are said to have a sticky nature at the temperature and pressure of the inlet steam, and therefore attach themselves to the blading and entrain and attach other salts as well. Professor Straub showed by laboratory tests that, in high-temperature steam, sodium hydroxide as well as tri-sodium phosphate and sodium silicate cause deposits, but that sodium chloride and sodium sulphate do not. He stated that the deposit in the case of tri-sodium phosphate and sodium silicate is due to the presence of sodium hydroxide in these salts and proved this statement by showing that carbon dioxide mixed with the steam ahead of the superheater prevented the deposits.

This picture presented by Professor Straub is very clear, but when the theory is applied to explain the deposits found in actual turbines certain discrepancies seem to appear. It is difficult, at least for the writer, to explain why in some turbines soluble deposits with a high percentage of sodium hydroxide are found, while in others insoluble deposits with a high percentage of silica occur. The explanation suggested by Tray,[10] that the silica deposits are laid down as sodium hydroxide and silica, and that due to operating conditions the sodium hydroxide is washed out, is hard to visualize. Sometimes there are found silica deposits which it is fairly certain have not been subjected to moist steam. Also, it would seem that if Tray's explanation is correct the resultant silica deposit would be porous and friable. The silica deposits, however, are not porous but form a hard impervious coating which clings firmly to the blades.

[8] "Descaling Steam Turbines," by Entriken, Ind. and Eng. Chem., vol. 30, p. 1279, Nov. 1938.

[9] "Salt and Silica Scale Deposits in Steam Turbines," by Goerke, Electrizitätswirkshaft. vol. 38, p. 61, Sept. 1939.

[10] "Boiler Operation as it Affects Prime Movers," by Tray, Mech. Eng., vol. 60, p. 475, June 1938.

From the pressures and temperatures existing in the turbine stages and Professor Straub's curves, the concentration of the sodium hydroxide solutions can be calculated throughout the turbine. These calculations give concentrations of 85 to 90 per cent sodium hydroxide in the incoming steam and show a very slow decrease in concentration as the steam flows through the first five or six stages and then a more rapid decrease to the dew point. The concentration and its change are practically the same, however, for both the turbines which have soluble deposits and those which have silica deposits. The concentration alone does not seem therefore to be a criterion for the kind or location of deposits formed. Perhaps for each concentration the physical characteristics of the solutions are different for different temperatures and pressures.

As noted previously, the deposit is most likely to occur in the nozzle throats and other regions of low pressure in the steam passages. This characteristic seems more pronounced than would be expected if the deposition of the salt were due purely to its sticky nature. Small liquid or solid particles would tend to concentrate on the concave surfaces and in the end whirls or secondary flow portions of the nozzle. Some deposit is found on the concave surfaces , but relatively much less than would be expected if the deposit were laid down as a sticky liquid and solid particles.

The occurrence of turbine deposits is similar to the formation of ice on airplane wings, insofar as both take place in regions of low pressure. This might suggest that the deposition of salts in the turbine is a saturation phenomenon.

The theory that salts are precipitated out of solution in high-pressure steam to form deposits is proposed by Dr. F. Spillner.[11] Spillner's tests show that sodium and potassium chloride and sodium hydroxide are soluble in saturated and superheated steam in rather surprising amounts, and that sodium sulphate is soluble also, but very much less so. The solubility of these salts increases very rapidly with pressure. Almost 2 ppm.* of sodium hydroxide will dissolve at 1200 psi. and the amount of hydrate is increased 10 fold for each 1000 psi. increase in pressure. Sodium and potassium chloride are soluble in like amounts, but sodium sulphate is less soluble in the ratio of 50 to 1.

The tests show also that the amount of the salt dissolved in the steam is independent of the salt concentration in the boiler above a concentration of about 100 ppm. From 100 ppm. boiler-water concentration to 20 ppm., the solubility in the steam decreases about 20 fold.

Although Dr. Spillner's results show that 2 ppm. of sodium hydroxide are soluble in 1200-psi. steam, no such amount of solid is found in the steam from a modern 1200-psi. boiler. This may of course be due to the fact that the steam is washed with the feed water which contains much less sodium hydroxide than the boiler water. However, if salts are carried into the superheater mechanically in the water droplets, and the salts are soluble in the steam, they still may be dissolved in the superheater and carried to the turbine in solution form.

The tests show that sodium chloride is more soluble in superheated steam than in saturated steam below a pressure of approximately 2000 psi. and less so above this pressure. If this characteristic is found to exist in the other salts also, some will be precipitated in the superheater and others will be carried to the turbine. The kind and amount of the selection will depend on the steam pressure.

The questions raised in the foregoing consideration of Professor Straub's theory of blade deposits can readily be answered if it is assumed that the

[11] "High Pressure Steam as a Solvent," by Spillner, Die Chemische Fabrik, vol. 13, p. 405, Nov. 1940.

* Parts per million.

salts actually are carried in solution in the steam. For example, the deposits would be expected to occur more thickly at points of rapid pressure drop because the solubility of the salts decreases with pressure. And in high-pressure turbines the increased solubility of the salts at the higher pressures keeps them in the steam until a later stage in the turbine is reached. The manner of formation of the silvery gray silica deposits, however, is still not clear.

Whatever the true explanation of how turbine deposits are laid down, the most effective method of preventing them, within the writer's experience, seems to be the sodium sulphate treatment suggested by Straub.[1] This is the method that the State Line Plant of the Chicago District Electric Generating Corporation has employed with great success, as mentioned earlier in this article. A brief outline of the experience which this plant has had in preventing deposit in turbines is contained in the following quotation from a communication to the writer by Mr. Frank Umbehocker, plant engineer:

"From 1930 to 1934 we attempted to control turbine-blade deposits by reducing carryover mainly through reducing the dissolved solids in the boiler water. That this does not work you can bear witness by recalling what the high-pressure turbine looked like when opened in 1931.† Even though we limited our boiler-water concentration to less than 350 ppm. and had steam with a specific resistance of 1,000,000 ohms or more (less than 0.5-ppm. carryover) we still had trouble with fouling.

"In the ASME paper‡ you will find some plant data. This was State Line Unit One. You will note that at the time the paper was published the progress reported was that we had increased the periods between washing of the high-pressure turbine. Later we eliminated washing completely by careful maintenance of a 5:1 ratio of Na_2SO_4 to NaOH. We found it necessary to check boiler waters daily, whereas in 1935 it was being done once per week. We also found that the usual turbidity method of sulfate determination was very unreliable, giving errors as high as 100 per cent. This was one reason why we did not eliminate washing completely until 1937. For a time we determined the sulfate gravimetrically to insure the 5:1 ratio, but are now using the di-sodium tetra-hydroxy-quinone method which is quite reliable."

Obviously, this or any other control practice that prevents the formation of deposits on turbine nozzles and buckets, and thus continuously provides the resulting benefits is more valuable than a washing method that provides the benefits only periodically.

SUPPRESSION OF NOISE IN OIL ENGINES.

The following article, first published in the "Sulzer Technical Review", is here reprinted from The British Motor Ship, August, 1942. A comprehensive examination is made of the problem of silencing Diesel machinery in ships.

In shipbuilding, the Diesel engine is being more and more adopted for the propulsion of passenger liners, where the horsepower, in relation to the size of the ship, is often very considerable. Since the weight of the engine per horsepower developed has to be much reduced, the rigid and heavy construction which helps to damp the noise is to a large extent eliminated.

† Illustrated in Figures 1 and 2 of this article.
‡ The A. S. M. E. paper referred to in footnote 1.

Therefore here too the endeavor to reduce noise has become an extremely acute problem.

The annoyance caused by noise depends not only on its intensity, i.e., its pressure amplitudes, but also on its character. Every noise is a composition of individual tones of different frequency and wave length, which may also differ greatly with respect to their loudness. Any endeavor to reduce noise effectively presupposes, therefore, an exact knowledge of its composition, as the damping methods hitherto known are not all equally efficacious for every frequency. The frequency-spectrometer allows every noise to be analyzed and its frequency-spectrum to be obtained (Figure 1). There the different

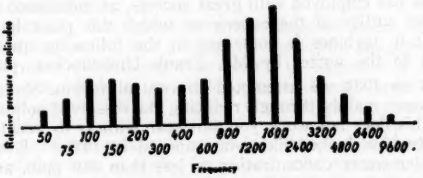


FIGURE 1.—FREQUENCY SPECTRUM OF THE EXHAUST NOISE FROM AN INTERNAL-COMBUSTION ENGINE.

tones appear, arranged in the order of their frequencies as vertical lines, the length of each being a measure of the intensity or pressure amplitude of the respective tone. Frequencies from 20 to over 10,000 Hertz are perceived as tones by the human ear, but all of them do not cause annoyance to the same extent, even when equally loud.

It is well known that high frequencies are much more liable to be annoying than low ones, and this has to be taken into consideration when building silencing devices. With quite low frequencies it will require great loudness to cause annoyance, and then it will be not so much the tone as the pressure fluctuations which produce disturbance in the ear.

If noise has to be successfully suppressed in Diesel engine installations, the different causes of production of the various noises must be sought for one by one and it has to be considered whether the formation of noise can be hindered or restrained by appropriate means, or if we must confine ourselves to checking its propagation and to reducing its pressure amplitudes in the direction of propagation.

Knocking in Diesel engines is a result of ignition lag, i.e., of the time elapsing from the beginning of injection until ignition occurs. The longer this time, the more fuel will be ready for combustion at the moment of ignition, and then it will burn practically instantaneously, causing a sudden rise of pressure. Therefore all methods which reduce ignition lag are suitable for diminishing the noise of ignition. The first remedy thought of will be to employ fuels easy to ignite, but this may not always be possible since the kind of fuel available is often determined by local conditions or economical reasons. As a rule this will generally not allow the use of suitable additions to the fuels for raising the ignitability. The most simple method is to adopt higher compression pressures, but this cannot be adopted indiscriminately, since a high compression pressure may give rise to high ignition pressures, which exceed the admissible limits. The noise of ignition can also be diminished by altering the cross-sections of the injection nozzles in order to obtain finer fuel jets. But as this reduces their power of penetrating the air in the cylinders, there is less intimate mixing of the fuel with the air, thus

reducing the effective output of the engine. If the fuels used are especially slow to ignite, tar oil for example, incandescent plates are fitted to the piston in order to obtain a local rise in temperature and thus facilitate ignition. Finally the quantity of fuel injected during the ignition lag period can be diminished by adopting slower injection.

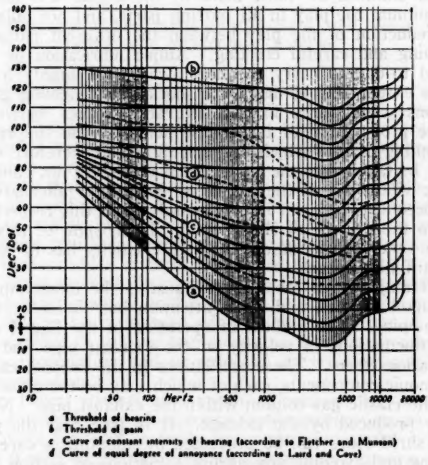
Valve Gear.

The noise caused by the movement of the valve gear is by no means inconsiderable, and it is due to the amount of play which is requisite to ensure certain closing of the valves, but also permits the various parts of the mechanism to knock against each other. This play will be kept down to an irreducible minimum, and the cams must be provided with suitably designed lifting surfaces in order that the play may disappear gradually and not suddenly. Since all shocks give rise to vibrations, which in their turn may cause noise, the pivots and fulcrums of the valve-gear members must be fixed, as far as possible, to big masses of low natural frequency; they should never be fixed to lighter parts, such as sheet-metal casings. Where the valve gear is operated by toothed wheels, it will be found advantageous to use wheels with oblique teeth.

However, to diminish the noise made by the engine it is necessary to reduce to a mimimum the play in all moving parts, and not only in the valve gear. This reduction of the play between the different parts requires accurate machining and careful erecting. Ample lubrication by forced circulation, as used to-day to carry the heat away from highly loaded engines, will to a large extent help to damp the shock in the driving gear, owing to the oil cushions which form in the clearance spaces between pins and bearings. The propagation of the engine noise can be suppressed to some degree by adopting casings which are dust-tight and therefore tightly closed, but care must be taken that the casings and doors are not caused to vibrate because of the vibrations of the engine or because of alternating pressures, or they may become sources of noise themselves. In this respect cast casings are superior to those made of sheet metal. If, in spite of this, sheet metal is chosen from other considerations, its high elasticity has to be counteracted by applying stiffening means.

In the great majority of cases the exhaust is the most important source of noise in Diesel engines, and great attention has therefore to be paid to its acoustic damping. The method of operation of the Diesel engine necessarily causes fluctuations in velocity in the exhaust pipe and consequently pressure oscillations there. These oscillations in the exhaust can be resolved into their harmonic components, each of which may be the cause of resonance vibrations in the elastic gas column within the exhaust pipe. Noise is therefore inevitably produced by the exhaust. It is true that the production of eddies and of shrill tones can be prevented by adopting a carefully planned system of piping and avoiding any abrupt variations of section or projecting edges. Nevertheless the pressure fluctuations will still remain and they are the chief cause of exhaust noise. First of all, a rigid construction must be adopted in order to counteract the tendency of the exhaust pipe to be set into vibration by the fluctuating internal pressure; otherwise the pipe may itself become a source of noise. Here also, as in the casings, cast iron is better than sheet iron. Especially slow to vibrate are double-walled cast pipes with cooling-water jackets. Nevertheless, owing to present-day considerations respecting weight and price, many exhaust pipes are built of sheet iron. The lagging adopted for thermal insulation—asbestos, glass wool, etc.—helps greatly to hinder their vibrating. But when the exhaust pipe is constructed by welding, the bends ought to be specially reinforced, since they are the parts most liable to be set into vibration.

The frequency spectrum of the exhaust noise shows broad frequency bands, that is to say, the corresponding tones of very different wave lengths are united to form outstanding groups of tones, which themselves in turn are distributed over the whole range of perceptible tones. Besides that, the different tones occur also with varying periodicity, so that the whole problem of exhaust noise must be described as being extremely complicated. It is therefore impossible to indicate any universal means for damping the exhaust noises, since the efficacy of the methods known to-day is not the same for different frequencies. By taking the frequency spectrum of the exhaust noise, knowledge is of course obtained of the ranges of specially outstanding frequencies, and the form of the silencer is chosen correspondingly, taking also into consideration that the tones of long wave length spread spherically, i.e., in all directions, whilst the higher tones are propagated mainly in the direction of the gas flow. But when damping devices are installed in the exhaust-pipe system, they cause the form of the oscillating gas column to change, and the dominating frequencies to be displaced. There is consequently a risk of obtaining results contrary to those desired, since with the modified disposition of the exhaust pipe certain tonal regions may be quite considerably amplified by resonance, regions which were of little or no im-



a Threshold of hearing
b Threshold of pain
c Curve of constant intensity of hearing (according to Fletcher and Munson)
d Curve of equal degree of annoyance (according to Laird and Coye)

FIGURE 2—FIELD OF HEARING.

portance with the original layout without any silencing devices. It is therefore impossible to determine the construction of an effective silencer in advance merely by analyzing the exhaust noise when working without a silencer, and then trying to reduce the tones of especially big sound-pressure amplitudes thus determined. The experimental investigation of silencers consists not in making measurements with and without the silencers, but in determining the acoustic pressure before and after them; only then the influence of various constructive arrangements within the silencer can be clearly established. As a source of noise, it will be found best to use a loudspeaker built into the inlet end of the exhaust pipe and two microphones, one placed in the pipe

before the silencer, and the other after it. The instruments have to be connected in such a way that the ratio of the acoustic pressures can be read direct on the indicating instrument (Figure 3). In practice, however, care must be taken not to assume that the results of such experiments are equally valid when the Diesel engine is running, since what appears to be good experimentally may prove to be quite impossible in the Diesel plant owing to the resistance there being too great.

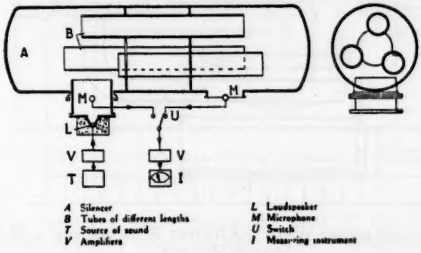


FIGURE 3.—ARRANGEMENT WITH BUILT-IN INTERFERENCE TUBES FOR TESTING A SILENCER COLD.

EFFECTS OF THROTTLING.

In principle it may be said that throttling, i.e., the temporary conversion of pressure energy into kinetic energy, after which the kinetic energy is converted by turbulence and internal friction into heat, damps the noise of the exhaust, since the intensity of the noise depends on the amplitudes of the pressure waves. But any resistance to flow in the exhaust pipe is equivalent to a loss of output; in two-cycle engines it may even cause considerable trouble with the scavenging and can therefore be adopted only with great care. Nozzles fixed at the end of exhaust pipes damp the pressure waves by conversion into increased kinetic energy. The damping effect is thereby assisted by the reduction in cross-section of the source of sound. On motorships such exhaust nozzles are commonly applied; their outlet section is rectangular, with the longer axis in the direction of motion of the ship. Because of this, the air current created by the movement of the ship offers less resistance to the exhaust flow, the gas jet remains longer intact and the tones of high frequency are carried farther away from the ship. By covering the funnel around the nozzle with perforated sheet iron, stiffened against vibrations, a considerable part of the tones of low frequency, propagated spherically in all directions, can be suppressed. However, experience has shown that the nozzle must project above the edge of the funnel, otherwise the funnel itself will easily be set vibrating (Figure 4). An exhaust-gas turbine installed in the exhaust pipe has a similar effect to throttling, although of course it converts the pressure energy into useful work. The turbine guide apparatus increases the velocity at the cost of the pressure, and in the wheel energy is withdrawn from the gases by changing their direction of flow, the gas stream being also finely divided up. Therefore the exhaust noise after an exhaust gas turbine is insignificant.

Since sound consists of waves, it is also possible to damp it by compensation, different sound waves being made to act on each other with phase displacement. Where single tones have to be suppressed, care must be taken to have the superimposed oscillations of the same frequency, otherwise beats

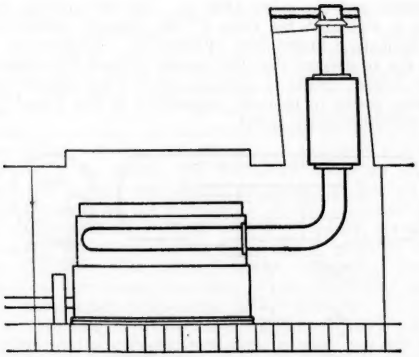


FIGURE 4.—SILENCER WITH EXHAUST NOZZLES FOR A MARINE DIESEL ENGINE.

may be caused, which are still more disagreeable in their effect than the original tones. But with the complex character of the exhaust noises, the risk of these beats occurring is not very great. Naturally compensation is most effective with a phase displacement of half a wave length, so that the peak of one wave will coincide with the trough of the other. But that kind of compensation can only be realized if the individual frequencies to be damped are exactly known and remain stable. Moreover, the technical application of compensation by superimposition of sound waves with a phase displacement of half a wave length is limited to high tones, because the necessary apparatus are of dimensions equivalent to half the wave length of the tone to be suppressed. Nevertheless, in this manner it has been found possible in special cases to eliminate successfully a particularly predominant high tone. To apply this method, the gas stream is divided into two, so that

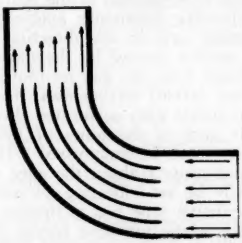


FIGURE 5.—COMPENSATION SILENCER WITH MULTIPLE SUB-DIVISIONS.

the path of one is half a wave longer than the path of the other, thereby producing the desired phase displacement when the two streams again unite.

The efficacy of the silencer shown in Figure 5 depends rather on chance. The division into many separate branches increases friction considerably, and consequently diminishes the acoustic energy, but causes a corresponding higher resistance. The unequal length of the different paths causes the phase

displacement, but it becomes effective only when the differences in length are of the order of a half wave length. Therefore this arrangement is applicable only to the damping of very high tones. Since the exhaust noise contains waves of different length and different phase displacement, also a silencer as shown in Figure 6 will reduce the sound pressure by compensa-

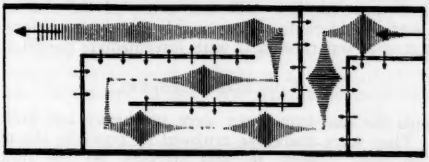


FIGURE 6.—COMPENSATION SILENCER WITH SEVERAL CHANGES OF DIRECTION BY MEANS OF PERFORATED PLATES.

tion, unless by an unfortunate chance the peak of the waves are added together, instead of partly cancelling each other. Generally, however, there will be differences of pressure on both sides of the perforated plates.

If the exhaust pipe is lined internally with a soft and porous material, the pressure waves are not reflected from the walls but are absorbed by them. As long as that lining remains soft and porous, it has a good silencing effect, but as soon as the pores become full of combustion residues, the absorbing effect will cease. For this reason such a construction cannot be recommended for damping noise from the exhaust. The case of silencers in the form of filters is analogous. The gases there flow through a porous filter material and lose much of the sound energy because of the great friction. Even when clean, the filter already causes a high resistance which is unfavorable to the engine, and as the filter becomes more and more obstructed, the engine may have to be stopped.

The type of exhaust silencer most generally used is that in which the cross-sections are varied; the bigger the ratio of the cross-sections, the better will be the damping effect. The abrupt alterations in cross-section when passing from pipe to silencer and again from silencer to pipe, both contribute to the damping effect. According to theoretical investigations the length of the apparatus does not influence the efficacy of the damping, but it does determine which frequencies will be particularly damped. If low tones have to be suppressed, the silencer must be long. This consideration, combined with the influence of the ratios of the sections, leads to a very voluminous silencer for a big, slow-running engine, because of the great amount of sound energy emitted at low frequencies. The full effect of the changes in cross-section can only be obtained when the silencer is fitted between rather long pipes. That involves the risk that the gas column in the end pipe may be excited to resonance, thus rendering the effect of the silencer illusory. If this is the case, the most simple remedy will be to alter the length of the pipe, if local conditions allow of this being done, otherwise a nozzle has to be installed at the end of it. In such cases, however, the installation of a second silencer would prove to be most effective, since it would not only prevent resonance in the end pipe, but also increase the damping effect. As already mentioned, the efficacy of silencing depends only on the ratio of the cross-sections and the number of times they alter, and not on the length of the silencer; thus it is natural to subdivide the silencer by means of partition walls. However, since the different chambers are then not connected by pipe pieces of a certain length, the increase in the damping effect is not

proportional to the number of subdivisions. With each shortening of the exhaust silencer the frequency band in which the silencing is most effective is displaced upwards into higher tones. Therefore it was quite natural that, as the speed of engines steadily increased, the design of silencers should develop from the single-chamber type into silencers divided into several separate chambers. The original silencers were built as big, one-piece, cast-iron vessels which, because of their stiffness, excluded the risk of vibration, but their weight was considerable; consequently it was soon found necessary to build them in several separate parts with intermediate partitions of sheet iron.

Concrete Silencers.

For big plants the cast-iron tanks were too heavy and bulky, and difficult to transport. They were therefore replaced by concrete chambers that could be built up on site. Owing to heat stresses, on the other hand, these chambers had to be built with thin walls, but since they had to withstand pressure waves caused by possible explosions in the exhaust, a strong reinforcement was requisite, and that meant very expensive constructions. In order to reduce the amount of reinforcements and hence the cost, an attempt was made to protect the walls by providing baffles at the inlet to break the pressure of the waves (Figure 7) and by fitting spring-loaded safety covers.

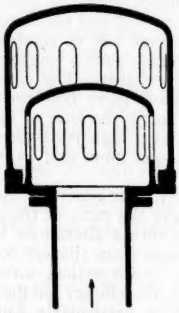


Figure 7.—Arrangement for Dispersing the Pressure Waves in Exhaust Pipes.

When the concrete chambers were constructed underground any cracks that occurred were rendered harmless and at the same time a certain cooling effect was obtained. Since Diesel plants are often erected in very remote places, where it is not certain that these reinforced concrete chambers can be properly constructed, silencers supplied with the engines are in many cases still given the preference. An endeavor was made to keep the volume of these new cast-iron silencers within reasonable limits without diminishing their damping effect, by fitting them internally with special baffle apparatus, and particularly with interference tubes (Figure 3). Nevertheless in big Diesel installations they attained such dimensions and weights that transport requirements made it indispensable to build the silencers of sheet metal. By adopting appropriate insulation, the vibrations in these tanks can also be prevented.

In proportion as the output of Diesel engines installed in ships and locomotives increased, it became more and more difficult to install a sufficiently effective silencer in the limited space available, until new ways were found by means of the so-called acoustic filter chains. The designation "chain" is characteristic for this apparatus, because they consist of a number of similar elements connected in series. They had their origin in electrical engineering, where they were introduced to damp certain groups of frequencies and are known as frequency filters. By analogy they have been transferred to acoustics for the same purposes.

This applies mainly to the "throttle-chain" and the "condenser-chain." In electrical terms a throttle chain consists of a series of oscillating circuits with throttles and condensers (Figure 8). In the acoustic filter-chain these

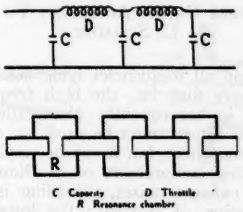


FIGURE 8.—ELECTRIC ACOUSTIC THROTTLE CHAIN ACCORDING TO A. KAUFFMANN AND U. SCHMIDT.

are replaced by resonance chambers and tubular pieces respectively. They have the peculiar property of damping all frequencies which are higher than twice their own natural frequency. Lower, that is to say deeper, frequencies can pass unhindered and in consequence those arrangements are called "deep-pass filters." As silencers they are generally built of two concentric pipes with partition walls in the outer one (Figure 9). The larger the resonance

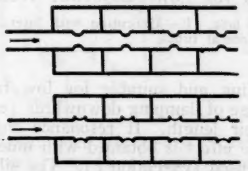


FIGURE 9.—SILENCER BUILT AS A LOW-FREQUENCY FILTER.

chamber, the lower extends the region of the damped frequencies, and in the limit case, it is possible to suppose that the outer jacket is omitted, and in this way the second group of acoustic filters is arrived at, namely the condenser chain (Figure 10). In the electric condenser chain the condensers are fitted in the transmission line, and the throttlers in the side lines, whilst acoustically speaking it consists of a number of Helmholtz resonators connected in series.

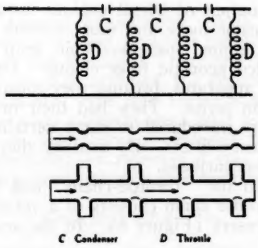


FIGURE 10.—CONDENSER CHAIN ACCORDING TO A. KAUFFMANN AND U. SCHMIDT.

Condenser chains damp all frequencies lying below half their natural frequency; those lying above that, i.e., the high frequencies, pass unhindered, and this arrangement is consequently also called a "high-pass filter." In that form a high-pass filter cannot be used for exhaust silencers, because of the connections to atmosphere, but it can be employed for damping the intake noise of reciprocating compressors or of Diesel engines. For damping the low frequencies of exhaust noises, throttling is used in addition to the already-mentioned variation of sections; the lower the frequencies to be damped, the more effective must the throttling be. Figure 11 shows a si-

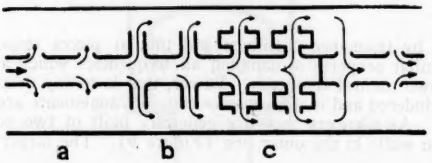


FIGURE 11.—SILENCER FOR LOW-FREQUENCY WORKING BY THROTTLING.

a.—Execution with single hole. b.—Execution with lateral tubes. c.—Execution with resonance chambers in the lateral tubes.

lencer based on throttling and suitable for low frequencies. The lateral branches extend the range of damping downwards (execution *b*), and they do so in proportion to their length. If resonance chambers are arranged in these branches, the same effect is obtained with much shorter branches than when smooth tubes are used (execution *c*), The silencing devices described above, and designated acoustic filters, are characterized by the small space they take up and, at least in the purely high and deep-pass types, by offering a minimum resistance to the flow.

In connection with the high-pass, its utilization has already been mentioned as silencer for the intake noise of reciprocating compressors or of scavenging pumps of two-cycle engines, and also of the cylinders of four-cycle engines. Nevertheless it is now very frequently replaced by Venturi nozzles, which can be arranged concentrically or alongside each other (Figure 12). If the conversion of the energy of the intake air takes place at the critical pressure ratio, so that the velocity of sound is attained in the smallest

cross-section, no sound waves can pass any longer from within to the outside quite independently of their frequency. But this high velocity causes a correspondingly great loss through friction and energy transformation, and the speeds generally used are kept as a rule much below the velocity of sound. Of course, the nozzle then no more acts as an absolute obstacle to sound waves issuing from within to atmosphere, but it still remains highly effective for damping low and medium frequencies. Air filters installed at the entry of the intake pipes, and filled with Raschig rings or, and still more effectively, with cocoanut fiber, have a good silencing effect, based on annihilation of the sound energy by friction. But, because of their high resistance, they are not used with turbo blowers as scavenging-air pumps or as super-

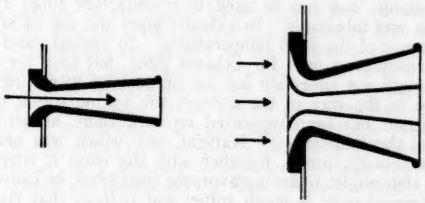


FIGURE 12.—SIMPLE AND SUBDIVIDED VENTURI TUBES.

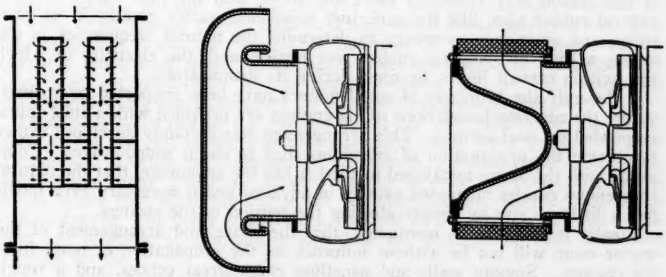


FIGURE 13.—LOW-FREQUENCY FILTER FOR SCAVENGING-AIR BLOWER. FIGURES 14 (CENTER) AND 15 (RIGHT).—ABSORPTION SILENCERS IN THE SUCTION BRANCHES OF SUPERCHARGING BLOWERS.

charging blowers in connection with Diesel engines. There the problem is mainly the damping of high tones, and, therefore, of high frequencies. Consequently, the "deep-pass" has been successfully applied for scavenging-air blowers (Figure 13), whereas the absorption dampers predominate for supercharging blowers (Figures 14, 15). The walls of the suction pipe are generally lined with felt as a sound-absorbing means. The sound waves leaving the turbine wheel cannot possibly get out without knocking first against the felt.

Engine Noise Travel.

The engine noises are communicated to the neighborhood not only by the surrounding air, but also by the vibrations of solid bodies, through the foundations and all rigid connections; insulation against the transmission of sound by solid bodies is of great importance in the endeavor to suppress noise. Not only in the air, but also in rigid bodies the sound waves represent elastic oscillations, and are damped in much the same way as all vibrations. Therefore all steps taken to check the transmission of vibrations to the surroundings, contribute also to diminish the nuisance caused by noise. The first of such measures is, for instance, to interrupt the rigid continuity of all pipes by inserting flexible tubing or stuffing boxes; this is rendered more easy nowadays, since the progress made in synthetic chemistry has made materials available that can be used to manufacture tubes resistant to the action of fuels and lubricants. In exhaust pipes the use of stuffing boxes is necessary because of the high temperatures. In special cases flexible metal tubing has already been used for exhaust pipes, but only for those of small diameter although not expressly for the purpose of insulating against noise. For insulation in the way of the ground, the engine frame or the concrete of the foundations has been supported on cork slabs, an air-gap being left open all round the foundations. Natural cork which was preferred at first because of its solidity, forms, together with the mass it supports, an oscillating system that might, under unfavorable conditions, be caused to vibrate in resonance. Pressed cork is much softer and reduces that risk, but without absolutely excluding it. Besides that, the elasticity of the pressed cork obtainable on the market differs so much, that the natural frequencies cannot possibly be determined in advance, and the weight of the mass which it supports, when caused to vibrate, may be sufficient to crush it completely. This is one reason why rubber is more and more used for such supports. But natural rubber also, like the cork just mentioned, varies greatly in its elasticity, and since it is necessary to determine the natural frequencies in advance, a kind of synthetic rubber was finally used, the elasticity of which can, within certain limits, be fixed during its manufacture.

Recently the bedplates of small motors have been supported on springs, whilst the concrete foundations of big engines are provided with a steel frame suspended on steel springs. This arrangement was certainly developed rather to prevent the propagation of vibrations than to damp sound waves. Compared with the above-mentioned method it has the advantage, that the natural frequencies can be calculated exactly in advance and if necessary even modified a little on site by simply altering the tension of the springs.

Finally it ought to be mentioned, that the shape and arrangement of the engine-room will not be without influence on the propagation of noise from the engines. Smooth walls and panelling cause great echoes, and a rough plaster surface is consequently preferable. In ships the walls of the engine-rooms are now covered with porous fiber slabs held in place and protected by netting or perforated sheet without impairing their sound-absorbing properties.

AXIAL VIBRATION OF DIESEL ENGINE CRANKSHAFTS.

Mr. Ralph Poole of the Metropolitan-Vickers Electrical Company, Ltd., is author of this paper reprinted from the February, 1942, Journal and Proceedings of the Institution of Mechanical Engineers. Mr. Poole describes investigations he has carried out to determine the magnitude of axial vibration of engine crankshafts, with the object of establishing that axial reson-

ance may occur within the normal running range of an engine. Formulae are offered for calculating axial deflection due to piston load and for axial loads. Empirical factors to be used in estimating axial stiffness were determined from a series of load-deflection tests carried out on a number of crankshafts. Mr. Poole suggests that the relation between axial deflections due to piston and axial loading determines the magnitude of the axial force set up by the piston loading.

Introduction.

Following an investigation of the vibration of a high-speed Diesel generator set, the opinion was formed that axial resonance might occur in engine crankshafts and that the stresses set up by this vibration might be of importance. The makers of the engine kindly placed the five engines in their power house at the disposal of the author and permitted him to carry out a series of tests. The theory and certain of the results were discussed with a number of designers, and later, two other cases of vibration in large marine engines were brought to the author's notice. In 1935, a typical record of the axial vibration of a high-speed Diesel engine was published in the author's contribution* to the discussion of a paper read before the Institution. As a result, further opportunities for testing were offered by several engine builders.

The present paper, in a different form, was originally submitted in 1937, since when Dr. Dorey has published† the result of an investigation of the axial vibration of one of the marine engines mentioned above. Many tests have been carried out by the author, and those included in the present paper are chosen as examples of the type of axial vibration which occurs in engine crankshafts.

A series of trials on engines of different makes cannot be carried out in the simple sequence possible in a laboratory. The author first found evidence of an axial critical speed during tests on a high-speed six-cylinder engine. The load tests on a crank unit similar to that used in this engine were carried out at a much later date. Subsequently load-deflexion and vibration tests were made on crankshafts lent by engine builders, and in one instance an opportunity was given to measure the axial stiffness of a crankcase.

Detailed descriptions of the engines tested are not included in the paper, and only in the case of the load-deflexion tests has it been possible to give dimensions. During the investigation, the author collected much data which suggest that many crankshaft failures are due to axial vibration, but experiments under controlled conditions are necessary before definite conclusions can be drawn. The paper is, therefore, confined to the problem of axial vibration and stiffness of crankshafts and does not attempt to deal with the stresses set up.

Method of Measuring Vibration.—Axial vibrations were measured by a Cambridge portable vibrograph fitted with a special extension consisting of a steel rod 0.25 inch in diameter, to which was brazed a steel ball 0.3 inch diameter. The ball was placed in a small hemispherical hollow in the center of the end of the shaft under test (Figure 1). The vibrograph traced a record upon a celluloid film by means of a stylus and required little or no attention during tests. An electric timing device marked the film at intervals of 1/10 second.

Torsional vibration was measured by a Cambridge universal vibrograph driven by a steel band, and the record was traced upon celluloid as de-

* Proc. I. Mech. E., 1935, vol. 131, p. 470.

† Trans. North East Coast Inst. Eng. and Shipbuilders, 1938-9, vol. 55, p. 203, "Strength of Marine Engine Shafting."

scribed above. A general view of the instrument is shown in Figure 2, Plate 1.

Engine Tests.

Single-cylinder Engine (*Engine No.* 1).—Axial vibration measurements were made on a single-cylinder Diesel engine capable of developing 100 HP. at 200 Rpm. The thrust block was at the flywheel end, and the engine was coupled to a d.c. generator by a belt on the flywheel rim. Two vibrographs were synchronized to record the axial movements of each end of the crankshaft simultaneously, in order to determine the relative phase of vibration.

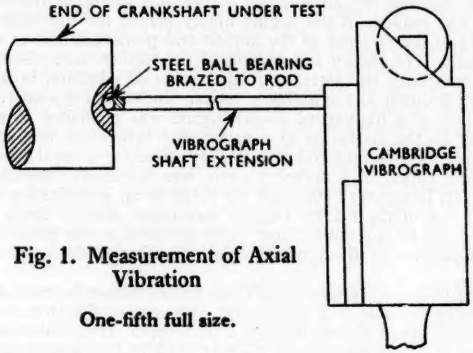


Fig. 1. Measurement of Axial Vibration

One-fifth full size.

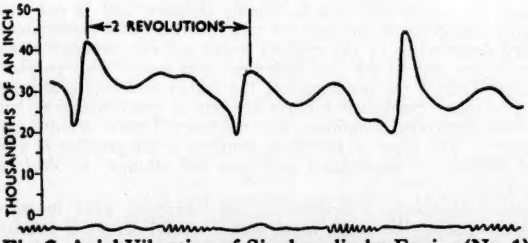


Fig. 3. Axial Vibration of Single-cylinder Engine (No. 1) Running Light at 200 r.p.m.

Upper curve, vibration at free end; lower curve, vibration at flywheel end.

The first set of records, taken with the engine running light at 200 Rpm., indicated that the free end of the crankshaft vibrated axially with an amplitude of approximately 0.024 inch, while the vibration at the flywheel end was negligible. The record taken at that end showed signs of a high-frequency vibration probably due to free vibrations of the flywheel itself at one of its natural frequencies (Figure 3).

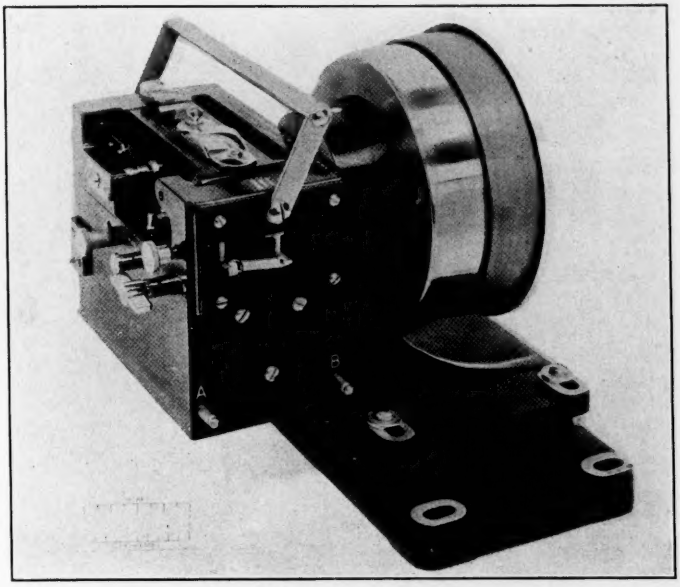


FIGURE 2.—CAMBRIDGE TORSIOGRAPH.

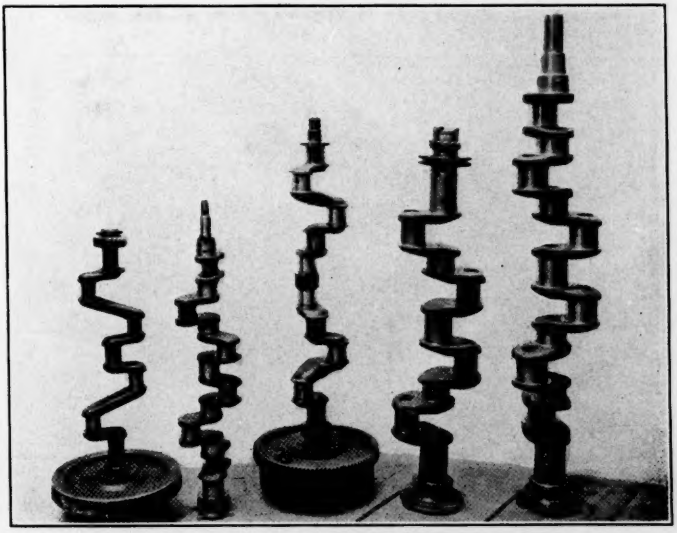


FIGURE 22.—CRANKSHAFTS ON WHICH TESTS WERE CARRIED OUT.

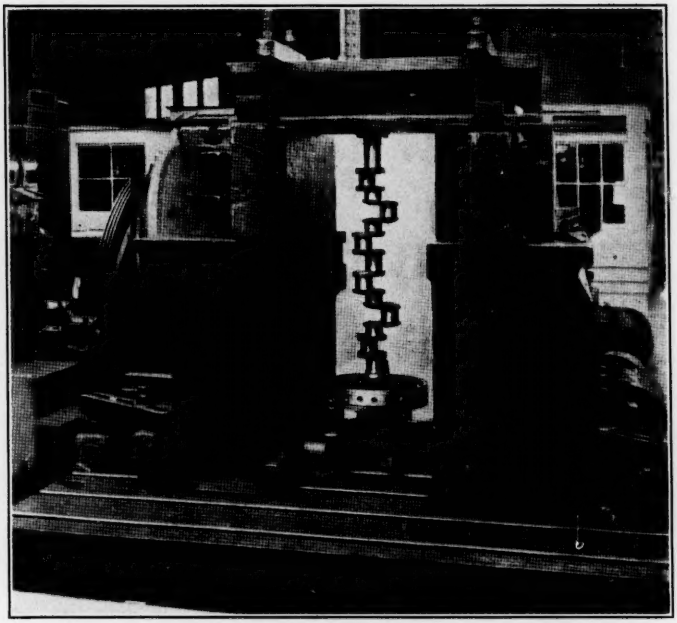


FIGURE 28.—ARRANGEMENT OF VIBRATION TEST ON CRANKSHAFT.

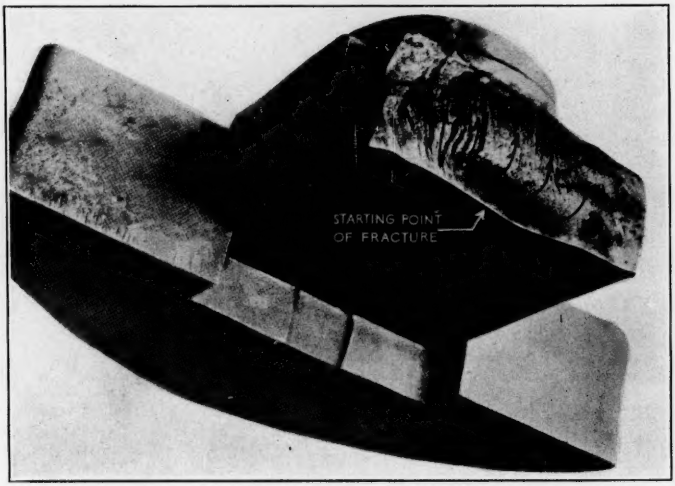


FIGURE 34.—FRACTURE OF CRANKSHAFT WEB DUE TO EXCESSIVE BENDING STRESSES.

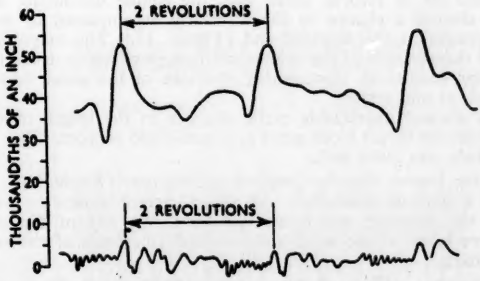


Fig. 4. Axial Vibration of Engine No. 1 at 200 r.p.m. when developing 80 h.p.

Upper curve, vibration at free end; lower curve, vibration at flywheel end.

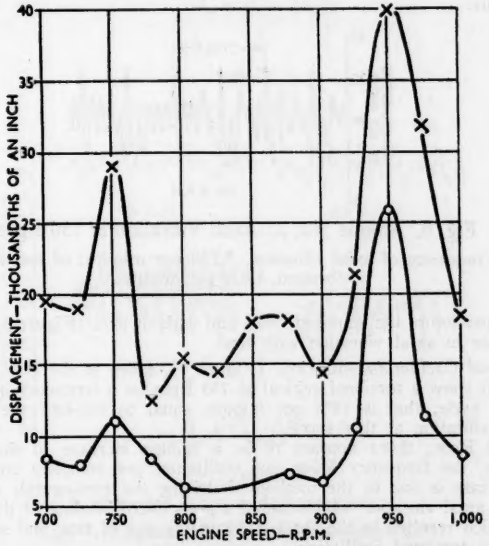


Fig. 5. Axial Vibration of Six-cylinder Rigidly Coupled Engine (No. 2), 700–1,000 r.p.m.

× Free end of engine. ○ Free end of generator

The second set of records taken with the engine developing 80 HP. at 200 Rpm., showed a change in the waveform accompanied by an increase in the movement at the flywheel end (Figure 4). The magnitude of the cyclic axial deformation of the crankshaft is approximately 0.018 inch under the particular conditions, representing the sum of the axial movements at the two ends at any instant.

The tests showed appreciable cyclic changes in the length of the crankshaft, of which the thrust block acted as a semi-rigid support. The maximum total amplitude was 0.025 inch.

Six-cylinder Engine Rigidly Coupled to Generator (Engine No. 2).—This engine had a built-up crankshaft. A direct-current generator was rigidly coupled to the flywheel, which was at the thrust end of the crankshaft. Records were taken of the axial and torsional vibrations of the engine and generator shafts.

Axial Vibrations.—Figure 5 shows the amplitude at the free ends of both engine and generator, plotted over a speed range from 700 Rpm. to 1000 Rpm., and peaks at 750 Rpm. and at 950 Rpm.

Figure 6 shows the vibration at 750 Rpm. It is of interest to note that there are ten oscillations per complete engine cycle, corresponding to a frequency of 3750 per minute, while the record at 950 Rpm. (Figure 7), shows twelve oscillations per cycle, corresponding to a frequency of 5700 per minute.

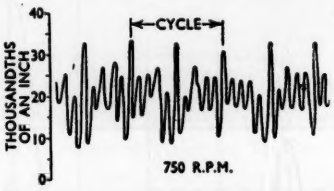


Fig. 6. Engine No. 2: Axial Vibration at 750 r.p.m.

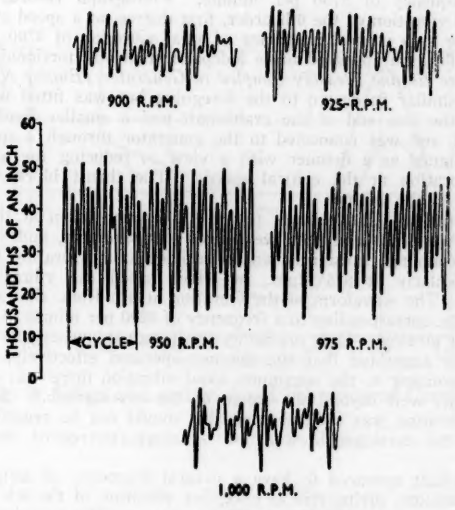
Frequency of axial vibration, 3,750 per minute; of torsional vibration, 1,875 per minute.

A comparison of the amplitude with and without load (Figure 8), indicates an increase in axial vibration with load.

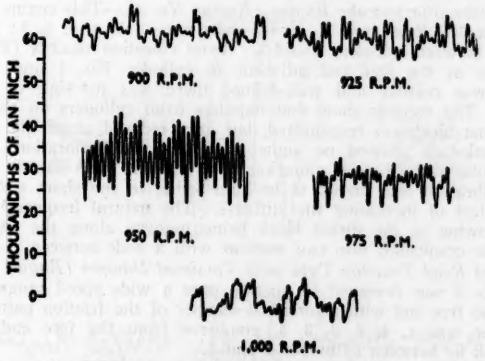
Torsional Oscillation.—Records (Figure 9), taken at the free end of the crankshaft show a torsional critical at 750 Rpm. at a frequency of 5 oscillations per cycle, that is 1875 per minute, equal to one-half the frequency of axial vibration at this speed.

At 950 Rpm., there appears to be a sudden increase in the torsional oscillation, the frequency being one oscillation per complete engine cycle. This increase is due to the method of driving the torsiograph and to the excessive axial vibration which caused a pronounced bending of the outboard crank which resulted in the shaft end running out of true and so gave the effect of a torsional oscillation.

The tests showed an increase in axial vibration when the engine ran at a torsional critical speed, the increase being most likely due to "spiraling" of the crankshaft. The natural frequency of the crankshaft showed a torsional critical vibration of the 2½ order at a speed of 750 Rpm., and a corresponding axial vibration at twice that frequency, i.e. of the 5th order



a Measured at free end of engine.



b Measured at free end of generator.

Fig. 7. Engine No. 2: Axial Vibration at 900–1,000 r.p.m.

giving a frequency of 3750 per minute. Vibrograph records showed a critical axial vibration of the 6th order, first degree, at a speed of 950 Rpm., corresponding to a natural frequency of axial vibration of 5700 per minute. This axial vibration appeared to be independent of any torsional oscillation.

Six-cylinder Engine Flexibly Coupled to Generator (Engine No. 3).—The engine was similar in design to the foregoing but was fitted with a main flywheel at the free end of the crankshaft and a smaller flywheel at the coupling end, and was connected to the generator through a special Bibby coupling designed as a detuner with a view to reducing the amplitude of torsional vibration at the critical speeds. The thrust block was at the coupling end of the engine.

Axial Vibration.—The records (Figure 10) were taken at the outboard end of the crankshaft and the free end of the generator, and showed that the flexible coupling transmitted an appreciable axial vibration to the generator, particularly at 875 Rpm., at which speed the vibration reached a maximum. The waveform of the vibration at 875 Rpm. shows 12 oscillations per cycle, corresponding to a frequency of 5250 per minute, (Figure 11).

Torsional Vibration.—Only negligibly small high-frequency vibrations were shown, which suggested that the detuner operated effectively, but at the speed corresponding to the maximum axial vibration there was one oscillation per cycle, well beyond the range of the torsiograph.* Slight high-frequency vibration was indicated but this should not be regarded as conclusive, as the torsiograph was too sensitive to record the complete fluctuations.

The crankshaft appeared to have a natural frequency of axial vibration of 5250 per minute, giving rise to excessive vibration of the 6th order, first degree, at a speed of 875 Rpm. There was no corresponding torsional oscillation of any important magnitude.

Six-cylinder Short-stroke Engine (Engine No. 4).—This engine developed 450 HP. at 500 Rpm., and its firing order was 1, 2, 4, 6, 5, 3; the thrust block was between cylinders 3 and 4. Axial vibration records (Figure 12) were taken at the free end adjacent to cylinder No. 1 and while the vibration was regular and well-defined there was no sign of a critical vibration. The records show that impulses from cylinders on the far side of the thrust block are transmitted, but at a reduced amplitude.

The crankshaft showed no signs of critical axial vibration, but there was a regular defined axial vibration at all speeds. The natural frequency of axial vibration was known to be high owing to the short stroke which had the effect of increasing the stiffness. The natural frequency was also increased owing to the thrust block being midway along the engine, thus dividing the crankshaft into two sections with a node between.

Engine of Road Traction Type with Torsional Damper (Engine No. 5).—Engine No. 5 was designed to operate over a wide speed range and was fitted at the free end with a torsional damper of the friction pattern. The firing order was 1, 4, 2, 6, 3, 5, numbered from the free end, and the thrust block as between cylinders 4 and 5.

Axial Vibration.—The records show 6 and 9 vibrations per cycle. Figure 13 gives the records taken when the engine was running with and without load respectively. Figure 14 is the record at 1110 Rpm.

Torsional Vibration.—Records (Figure 15) taken at the free end showed pronounced critical vibrations at 875 Rpm. and 1100 Rpm., corresponding to frequencies of 6 per cycle or 2625 per minute and 9 per cycle or 4950 per minute respectively.

* The torsiograph magnification had been adjusted to a maximum in order to record the small high-frequency vibrations.

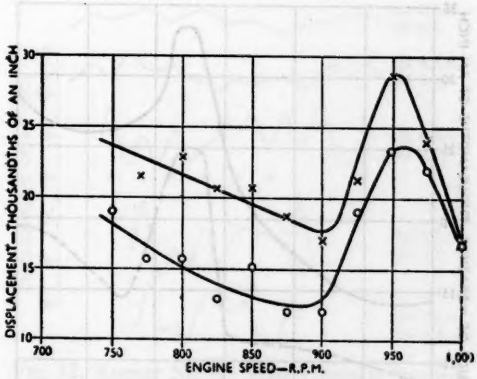


Fig. 8. Effect of Load on Axial Vibration of Crankshaft

o No load. × Part load.

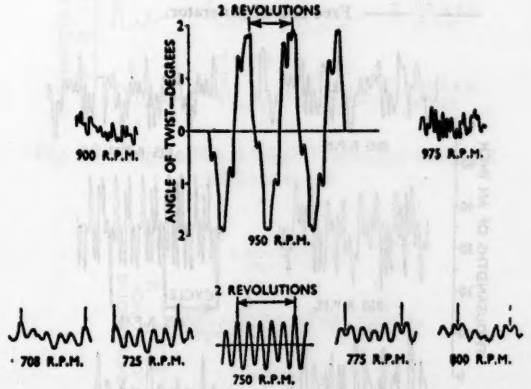


Fig. 9. Engine No. 2: Torsional Oscillation at Free End

For corresponding axial vibration, see Fig. 7.

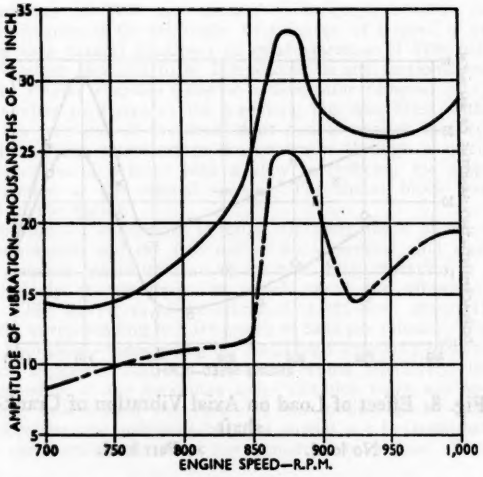


Fig. 10. Engine No. 3: Axial Vibration at Different Speeds

——————— Free end of shaft at main flywheel.
——— . ——— Free end of generator.

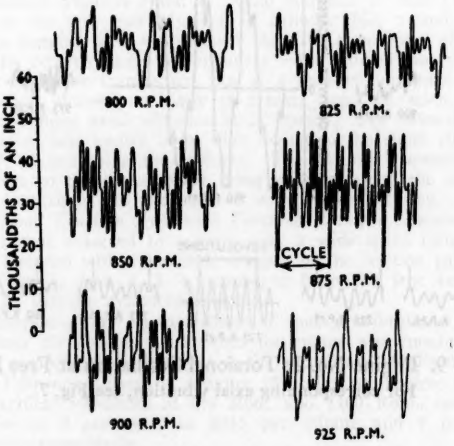


Fig. 11. Engine No. 3: Axial Vibrations at 800–925 r.p.m.

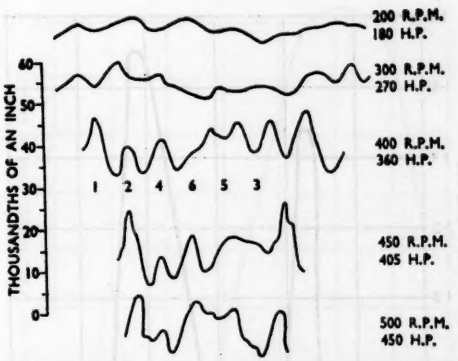


Fig. 12. Engine No. 4: Axial Vibration Records, 200–500 r.p.m.

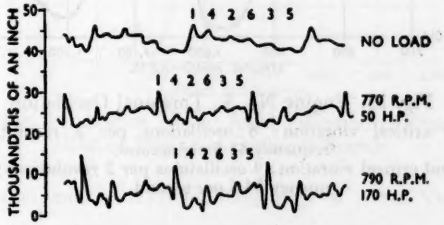


Fig. 13. Engine No. 5: Axial Vibration with and without Load

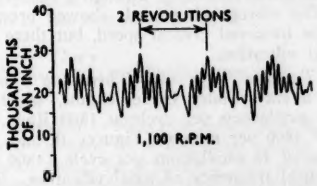


Fig. 14. Engine No. 5: Axial Vibration at Torsional Critical Speed

Analysis of these records shows 9 oscillations per 2 revolutions, corresponding to a torsional critical speed of 1,110 r.p.m.

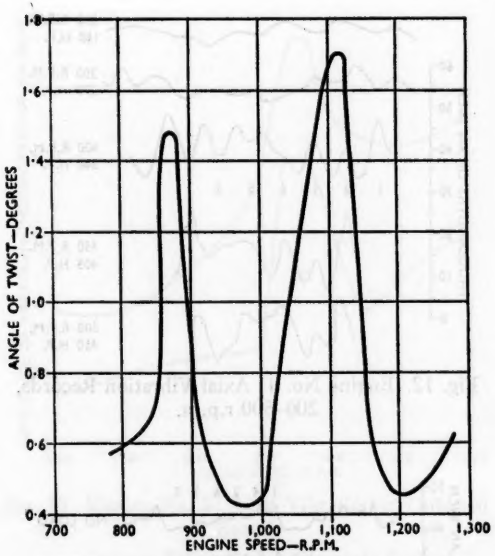


Fig. 15. Engine No. 5: Torsional Oscillation

First critical vibration: 6 oscillations per 2 revolutions; frequency 43·5 per second.
Second critical vibration: 9 oscillations per 2 revolutions; frequency 84·5 per second.

The crankshaft was extremely stiff and had a very high natural frequency responding only to high order impulses. Some detuning effect was also introduced by the thrust block between cylinders 4 and 5. The system became equivalent to two mechanically connected springs of different axial frequency, their connection being through a semi-rigid support, namely the thrust block. The vibrograph records showed pronounced increases in axial vibration at the torsional critical speed, but there was no evidence of an independent axial vibration.

Engine of Road Traction Type with Thrust at Coupling End (Engine No. 6).—This was a six-cylinder Diesel engine, which showed a critical axial vibration of 6 oscillations per cycle at 1600 Rpm., corresponding to a natural frequency of 4800 per minute (Figures 16 and 17). At 800 Rpm., there was a vibration of 12 oscillations per cycle (4800 per minute), which coincides with a natural frequency of axial vibration. Torsiograph records showed no corresponding torsional critical vibration.

The crankshaft had a natural frequency of axial vibration of 4800 per minute, which gave rise to a 6th order, first degree, axial virbation at 800 Rpm. and a 3rd order, first degree, axial vibration at a speed of 1600 Rpm. This axial vibration bore no relation to the torsional oscillation.

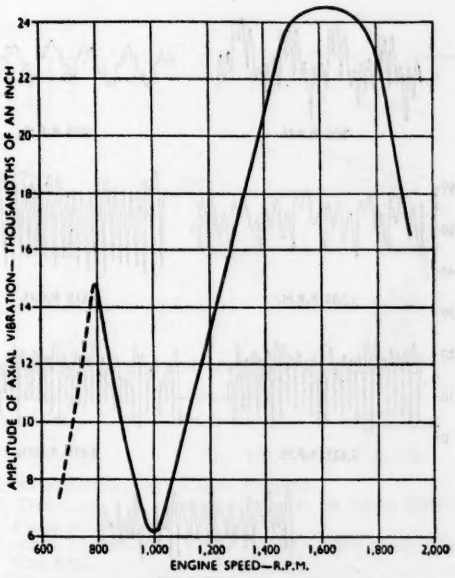


Fig. 16. Engine No. 6: Axial Vibration at Free End of Engine at Different Speeds

THE CALCULATION OF NATURAL FREQUENCY OF AXIAL VIBRATION.

Axial vibration may take place with the node at the thrust bearing, that is, as a first-degree vibration; or the node or nodes may fall along the crankshaft itself in the case of vibrations of higher degree. In the present tests there is no evidence of vibration higher than the first degree.

In vibration of the first degree, the crankshaft may be treated as a spring fixed at one end * and loaded at intervals from the node thus formed, when the natural frequency of the system is given by

$$187.8\sqrt{\frac{\Sigma wy}{\Sigma wy^2}} \text{ oscillations per min.}$$

where y is the axial deflexion in inches corresponding to the load w pounds.

This expression may be used for crankshafts having cranks of different weights at any distance along the crankshaft, but where the cranks are of equal weight and equally spaced, their total distributed weight W pounds can be replaced by an equivalent weight W/4.12 pounds concentrated at the free end.

* It should be noted that the natural frequency is modified by the clearance in the thrust bearing which allows free axial movement within certain limits. The reduction in the natural frequency of a system, permitted by freedom at the node, is dealt with in textbooks.

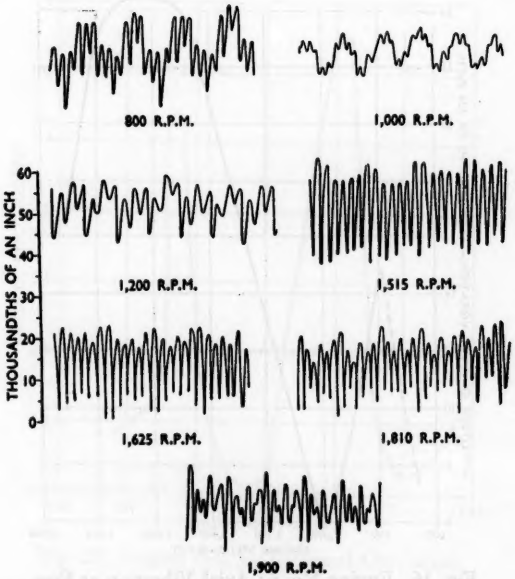


Fig. 17. Engine No. 6: Axial Vibration at 800–1,900 r.p.m.

THE AXIAL STIFFNESS OF A CRANK ELEMENT.

Axial Deflexion due to Radial Load on Crankpin.—It is required to find the axial shortening or reduction in length L of the crank element (Figure 18a) under the applied load at the crankpin. It is convenient to consider the loading symmetrical and concentrated on a half-crank element built in midway along the crankpin (Figure 18b) and equal to ½P. where P is the crankpin load in pounds.

In the equations given below the following notation is used, in addition to the conventional symbols M, l, E, and I which are generally adopted in equations for the bending, slope, and deflexion of beams:

2a	Crankpin length.	
b	Crank web length.	
2c	Journal length.	
I_p	Moment of inertia of crankpin.	
I_w	Moment of inertia of crank web.	
I_j	Moment of inertia of journal.	
1/A	Flexural rigidity of half-crankpin	$= EI_p/a$
1/B	Flexural rigidity of crank web	$= EI_w/b$
1/C	Flexural rigidity of half journal	$= EI_j/c$

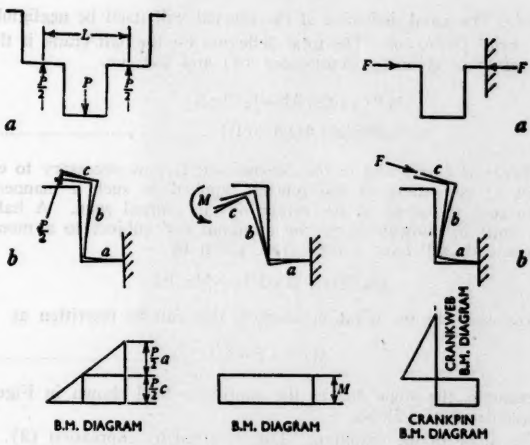


Fig. 18. Fig. 19. Fig. 20.

Fig. 18. Axial Stiffness of a Crank Element.
Fig. 19. Deflexions and Bending Moments set up in Half-crank Fixed at One End.
Fig. 20. Axial Stiffness of a Crank Element (Half-crank Fixed at One End).

Crankpin.—The slope Ml/EI, at the web end of the crankpin, due to the uniform bending moment ½Pc, is

$$\tfrac{1}{2}Pc \times \frac{a}{EI_p} = \tfrac{1}{2}PcA \quad \ldots\ldots (1)$$

and the slope $Wl^2/2EI$ due to the cantilever load ½P is

$$\frac{P}{4EI_p} \times a^2 = \tfrac{1}{4}PaA \quad \ldots\ldots (2)$$

The total slope at the end of the crankpin is the sum of the foregoing, i.e.

$$\tfrac{1}{4}P(a+2c)A \quad \ldots\ldots (3)$$

Crank Web.—From expression (3), the deflexion of the journal end of the crank web due to the slope of the crankpin is

$$\tfrac{1}{4}P(a+2c)Ab \quad \ldots\ldots (4)$$

The bending moment, which acts uniformly along the web, produces a deflexion $\frac{1}{2}M^2/EI$, equal to

$$\tfrac{1}{2}\left(\tfrac{1}{2}PC \times \frac{b^2}{EI_w}\right) = \tfrac{1}{4}PbcB \quad \ldots\ldots (5)$$

Journal.—The axial deflexion of the journal will itself be negligible.

Total Axial Deflexion.—The total deflexion for the half crank is therefore the sum of those given by expressions (4) and (5), i.e.

$$\tfrac{1}{4}P(a+2c)Ab+\tfrac{1}{4}PbcB$$
$$= \tfrac{1}{4}Pb[(a+2c)A+cB] \quad \ldots\ldots (6)$$

The Effect of Constraint in the Journal.—It is now necessary to examine the effect of constraint, at the journal, applied in such a manner as to reduce to zero the slope in the center of the journal span. A half-crank element, built in midway along the crankpin and subject to a moment M at the free end, will have a total slope given by

$$Ma/EI_p+Mb/EI_w+Mc/EI_j$$

at the free end. In the notation adopted, this can be rewritten as

$$M(A+B+C) \quad \ldots\ldots (7)$$

As an example, the slope due to the cantilever load shown in Figure 18b can be calculated as follows:—

(1) The slope in the crankpin. This is given by expression (3).

(2) The slope Ml/EI in the web due to bending. This is given by $\tfrac{1}{2}Pc(b/EI_w)$, which becomes

$$\tfrac{1}{2}PcB \quad \ldots\ldots (8)$$

(3) The slope $Wl^2/2EI$ in the half-journal due to the cantilever load. This is given by $\tfrac{1}{4}Pc^2/EI_j$, which becomes

$$\tfrac{1}{4}PcC \quad \ldots\ldots (9)$$

The total slope at the mid-section of the journal, due to reaction at the journal is therefore the sum of expressions (3), (8), and (9), i.e. $\tfrac{1}{4}P(a+2c)A+\tfrac{1}{2}PcB+\tfrac{1}{4}PcC$, which becomes

$$\tfrac{1}{4}P[(2A+2B+C)c+Aa] \quad \ldots\ldots (10)$$

For zero slope at the mid-section of the journal, the slope due to the end moment M (Figure 19) must be equal and opposite to that produced by the end load. Equating expression (7) and (10),

$$M(A+B+C) = \tfrac{1}{4}P[(2A+2B+C)c+Aa]$$

or
$$M = \tfrac{1}{4}P\left\{\frac{(2A+2B+C)c+Aa}{A+B+C}\right\} \quad \ldots\ldots (11)$$

Equation (11) thus shows that the end moment M is dependent upon the proportions of the crankshaft parts.

Axial Deflexion Produced by Constraint in the Journal.—The end moment M will produce an axial deflexion in the opposite direction to that of the deflexion caused by the end load. At the web end of the crankpin the slope is MA, which will set up a deflexion MAb at the journal end of the web. The deflexion of the crank web itself under the action of the same moment will be $\tfrac{1}{2}MBb$. The axial deflexion in the half-journal will be negligible,

TABLE 1. CRANKSHAFT D

Crankshaft		Dimensions, inches	Moment of inertia (ins.⁴)	1÷flexural rigidity (10^{-6})	
Number	Section				
No. 1; $E = 30 \times 10^6$ lb. per sq. in.	I	a 1·125 b 1·9 c 1·55	I_p 0·141 I_w 0·0668 I_j 0·2485	A 0·256 B 0·947 C 0·207	N
	II	a 2·0 b 3·8 c 2·0	I_p 0·141 I_w 0·348 I_j 0·141	A 0·473 B 0·363 C 0·473	
No. 2; $E = 30 \times 10^6$ lb. per sq. in.	—	a 0·913 b 2·0 c 1·063	I_p 0·4604 I_w 0·04068 I_j 0·7854	A 0·0662 B 1·638 C 0·0452	
No. 3; $E = 30 \times 10^6$ lb. per sq. in.	I	a 1·25 b 2·125 c 1·25	I_p 0·7854 I_w 0·0791 I_j 0·7854	A 0·053 B 0·895 C 0·053	N
	II	a 1·6 b 2·125 c 1·6	I_p 0·7854 I_w 0·6851 I_j 0·7854	A 0·068 B 0·125 C 0·068	
	III	a 2·875 b 2·125 c 1·25	I_p 0·7854 I_w 0·0790 I_j 0·7854	A 0·122 B 0·895 C 0·053	
No. 4; $E = 18 \times 10^6$ lb. per sq. in.	I	a 1·5 b 2·75 c 1·5	I_p 1·139 I_w 0·258 I_j 2·807	A 0·0732 B 0·592 C 0·0296	H
	II	a 1·5 b 2·75 c 2·0	I_p 1·139 I_w 0·258 I_j 2·807	A 0·0732 B 0·592 C 0·0394	
No. 5; $E = 18 \times 10^6$ lb. per sq. in.	I	a 1·5 b 2·75 c 1·5	I_p 3·5462 I_w 0·32 I_j 3·976	A 0·0234 B 0·476 C 0·0209	H
	II	a 1·5 b 2·75 c 2·0	I_p 3·5462 I_w 0·32 I_j 3·976	A 0·0234 B 0·476 C 0·0278	
No. 6; $E = 30 \times 10^6$ lb. per sq. in.	I	a 1·5 b 2·75 c 1·5	I_p 3·976 I_w 0·32 I_j 3·976	A 0·0125 B 0·2855 C 0·0125	A
	II	a 1·5 b 2·75 c 2·0	I_p 3·976 I_w 0·32 I_j 3·976	A 0·0125 B 0·2855 C 0·0167	

NKSHAFT DIMENSIONS AND CALCULATED DEFLEXIONS

flexural gidity 10⁻⁹)	Remarks	Estimated axial deflexion per 1,000 lb. axial load, thousandths of an inch		Estimated axial deflexion per 1,000 lb. piston load, thousandths of an inch		$\frac{\Delta_{TT}}{\Delta_{TT}}$	$\frac{\Delta_{TB}}{\Delta_{TB}}$
		No bearing constraint Δ_{TT}	Bearing constraint zero slope at journal Δ_{TB}	No bearing constraint Δ_{TT}	Bearing constraint; zero slope at journal Δ_{TB}		
0·256 0·947 0·207	No journal between cranks 1 and 2 or between cranks 3 and 4. Fig. 22, Plate 1	2·062	0·7	1·21	0·1477	0·586	0·211
0·473 0·363 0·473		8·58	3·855	3·386	0·8987	0·394	0·232
0·0662 1·638 0·0452	—	2·448	0·653	0·971	0·0272	0·396	0·0416
0·053 0·895 0·053	No journal between cranks 1 and 2, or 3 and 4, or 5 and 6. Spacer 1·5 inches long between cranks 3 and 4	1·592	0·460	0·7	0·036	0·44	0·0562
0·068 0·125 0·068		0·497	0·2011	0·2797	0·0579	0·563	0·288
0·122 0·895 0·053		1·905	0·640	0·943	0·1076	0·495	0·168
0·0732 0·592 0·0296	Hole 1¼ inches in diameter, ¼ inch from neutral axis in each crankpin. Centre journal 4 inches long	2·049	0·567	0·835	0·0565	0·406	0·0995
0·0732 0·592 0·0394		2·049	0·587	1·0908	0·0582	0·532	0·099
0·0234 0·476 0·0209	Hole 1¼ inches in diameter, ¼ inch from neutral axis in each crankpin. Centre journal 4 inches long	1·383	0·393	0·5633	0.0240	0·406	0·061
0·0234 0·476 0·0278		1·383	0·402	0·7432	0·0315	0·536	0·0785
0·0125 0·2855 0·0125	As No. 5 crankshaft, but without hole in crankpin	0·822	0·235	0·3335	0·0121	0·405	0·0514
0·0125 0·2855 0·0167		0·822	0·241	0·440	0·019	0·538	0·079

so the total axial deflexion produced by the end moment is given by the sum

$$MAb+\tfrac{1}{2}MBb = \tfrac{1}{2}Mb(2A+B) \quad \ldots\ldots (12)$$

Axial Deflexion of Crank Element: Load on Crankpin and Zero Slope at Journal.—The net deflexion $\triangle_P$ of the half-crank element, allowing for constraint at the journal is found by subtracting expression (12) from expression (6), i.e.

$$\triangle_P = \tfrac{1}{4}Pb[(a+2c)A+cB] - \tfrac{1}{2}Mb(2A+B) \quad \ldots\ldots (13)$$

When the journals are freely supported, $M = 0$; and when the constraint is such that there is no slope at the mid-section of the journal,

$$M = \tfrac{1}{4}P\left\{\frac{(2A+2B+C)c+Aa}{A+B+C}\right\}$$

Equation (13) holds for any value of the end moment M and is therefore applicable when the slope produced by it exceeds that caused by the end load or by journal reaction.

Axial Deflexion due to Axial Load.—The axial deflexion may be calculated by considering a crank element under the action of an axial force F acting along the journal (Figure 20a). If the crank is first assumed to be freely supported at the journals the problem can be represented as in Figure 20b by a half-crank element built in mid-way along the crankpin and having an axial force acting at the free end. The slope at the web end, due to the bending moment Fb acting uniformly along the crankpin, is

$$Fb\left(\frac{a}{EI_p}\right) = FbA \quad \ldots\ldots (14)$$

The deflexion at the journal end of the crank web due to the slope of the crankpin will be

$$Fb^2A \quad \ldots\ldots (15)$$

The axial deflexion of the crank web under the action of the cantilever load W is given by

$$Fb^3/3EI_w = \tfrac{1}{3}Fb^3B \quad \ldots\ldots (16)$$

The compression in the journal being negligible, no axial deflexion will be produced in the journal itself.

The total axial deflexion due to the axial load in the half-crank element is thus given by the sum of expressions (15) and (16), i.e.

$$Fb^2A+\tfrac{1}{3}Fb^3B = \tfrac{1}{3}Fb^2(3A+B) \quad \ldots\ldots (17)$$

The effect of constraint has already been considered, and equation (7) gives the slope at the journal due to an end moment M, the value of which can be calculated, for axial loads also, by equating the slope due to end moment to that produced by the end load. The total slope due to the axial load is the sum of the slope FbA in the crankpin and the slope in the web due to the cantilever load. The latter is given by

$$Fb^2/2EI_w = \tfrac{1}{2}FbB \quad \ldots\ldots (18)$$

The total slope at the journal is therefore

$$\tfrac{1}{2}Fb(2A+B) \qquad (19)$$

Equating expressions (7) and (19), for zero slope at the journal

$$M(A+B+C) = \tfrac{1}{2}Fb\,(2A+B)$$

or

$$M = \tfrac{1}{2}Fb\left(\frac{2A+B}{A+B+C}\right) \qquad (20)$$

Axial Deflexion with Constraint on the Journal.—Equation (12) shows the axial deflexion produced by the constraint to be equal to $\tfrac{1}{2}Mb(2A+B)$. The net deflexion $\triangle_F$, allowing for the constraint, is therefore found by subtracting expression (12) from expression (17), i.e.

$$\triangle_F = \tfrac{1}{3}Fb^2(3A+B) - \tfrac{1}{2}Mb(2A+B) \qquad (21)$$

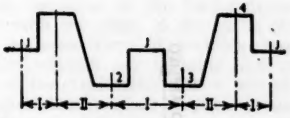
The Axial Force on a Crankshaft.

The piston loading, in addition to producing axial deformation in a crankshaft, also sets up an axial inertia force. In a simple crank element, fixed at one end and free to move axially at the other, the axial deflexion due to a piston load P may be denoted by Δ_P and that due to an axial force by $F\Delta_F$. If the crank element is now regarded as fixed (in the normal undeflected position) at the "free" end, the piston load P will set up an axial force $P\Delta_P/\Delta_F$, and this is the axial force required to produce an axial deflexion equal and opposite to that set up by the piston load when the "free" end of the crank element is once more free.

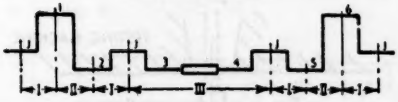
If a mass M acts at the free end of a crank element, no axial force will be set up if the piston load remains constant; but if the piston load varies, the inertia of the flywheel will oppose the axial motion of the crank, and an axial force, proportional to the rate of change of piston loading, will consequently be set up. The periodic variation in piston loading in an engine therefore produces a corresponding periodic axial inertia force in the crankshaft. The magnitude of the axial disturbing force on a crank element is thus equal to the force setting up an axial deflexion equal and opposite to that caused by the piston loading.

In equations (13) and (21) expressions have been developed for the deflexions Δ_P and Δ_F, in terms of the flexibility of the crankpin, crank web, and journal, from which the magnitude of the axial force can be calculated. In Table 1 typical values are given for the elements of the crankshafts shown in outline in Figure 21 and illustrated in Figure 22, Plate 1.

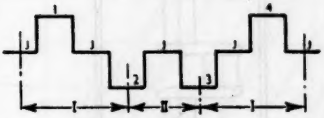
The magnitude of the resultant axial force can also be derived from axial vibration tests, when the masses are known and the vibration record has been analyzed. The importance of the higher harmonics is due to the force being proportional to $\omega^2 a$, where ω denotes the frequency of the component having an amplitude a. In some instances, pronounced second and third harmonics represent a rapid reversal of motion and are consequently evidence of the presence of large axial forces. As an example, a certain 1000 pound flywheel requires a force of 365 pounds to oscillate it through 1/1000 inch at a frequency of 60 per second, while at a frequency of 120 per second a maximum force of 1460 pounds is required to produce the same movement. Similarly, at 180 cycles per second the maximum force per 1/1000 inch displacement from the mean position is 3295 pounds.



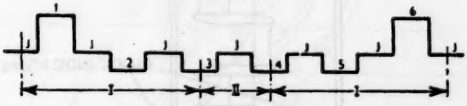
Crankshaft No. 1.



Crankshaft No. 3.



Crankshaft No. 4.



Crankshafts Nos. 5 and 6.

Fig. 21. Division of Crankshafts for Calculations

The arabic numerals on the diagrams represent cranks. The letter J represents journals. Roman numerals define the various sections.

The Axial Stiffness of Crankshafts.

The natural frequency of axial vibration of a crankshaft will depend upon its mass and its axial stiffness. The axial stiffness of a crank element has already been considered theoretically, and expressions have been derived for the bending moment set up by the supports. There is, however, considerable difficulty in applying the theory to practical cases. The problem is complicated by the fact that the crankpins and journals are cylindrical, while the web is rectangular in cross-section, and the "effective" or equivalent lengths of the crankpin, journal, and web respectively may differ appreciably from their physical dimensions. Moreover, the neutral axis is not a straight line normal to the plane of the crank.

Axial Load Tests on Crankshafts.—In order to find empirical factors which would enable the axial stiffness of a crankshaft to be determined by calculation, load-deflexion tests were carried out, by means of an Olsen testing machine, on a number of crankshafts having a wide range of proportions. The axial deflexions were measured by two indicators (Figure 23), and the mean of the two indicator readings was taken as the true deflexion.

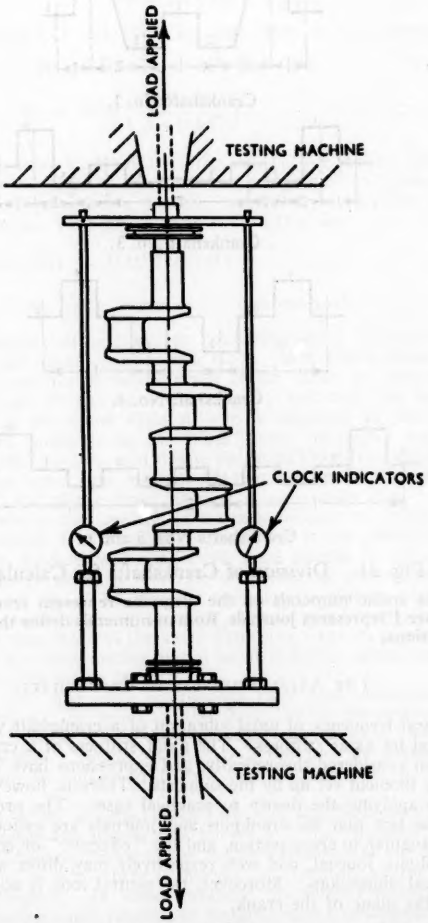


Fig. 23. Method of Measuring Deflexion of Crankshafts under Load.

Since an increase in the slope of the load-deflexion curve under compression, as compared with the slope of the curve obtained under tension, might be attributable to some form of buckling, tensile and compressive tests were made to find whether any buckling took place. By pure coincidence the first three of the crankshafts were of almost equal axial stiffness, although they were taken from engines of different make and size (Figure 22, Plate 1). The stiffnesses of the fourth and fifth crankshafts were almost twice those of the first three. Curves showing the results for all five crankshafts are given in Figure 24.

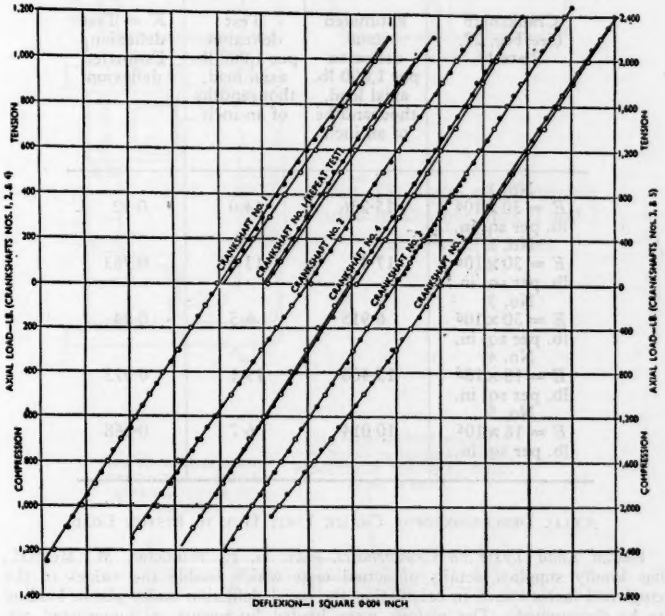


Fig. 24. Load-Deflexion Curves

● Increasing load.
○ Decreasing load.

Comparison between Measured and Calculated Deflexion.—In Table 2 the measured stiffness is compared with the calculated stiffness; it is assumed that the effective journal length is equal to the actual journal length plus the web thickness. Similarly the effective crankpin length is taken to be equal to the actual crankpin length plus the web thickness (Figure 21). The greatest discrepancy between test result and calculation occurs in the case of crankshaft No. 5, and is no doubt due to the fact that the crankpin and journal diameters overlap, so that the fibers of the material are continuous in this region, with the result that the bending moment caused by an axial

load is considerably reduced. The ratio of the deflexion obtained during test to the calculated deflexion is denoted by K; and the average value of K for these five crankshafts is 0.85; this figure can be regarded as an empirical coefficient which when applied to the calculated results, brings them closely into line with the test results.

TABLE 2. MEASURED STIFFNESSES AND CALCULATED STIFFNESSES

Crankshaft (see Fig. 22, Plate 1)	Estimated axial deflexion per 1,000 lb. axial load, thousandths of an inch	Test deflexion per 1,000 lb. axial load, thousandths of an inch	K = Test deflexion/ Estimated deflexion
No. 1 $E = 30 \times 10^6$ lb. per sq. in.	15·236	14·0	0·92
No. 2 $E = 30 \times 10^6$ lb. per sq. in.	17·4	13·6	0·783
No. 3 $E = 30 \times 10^6$ lb. per sq. in.	6·915	6·5	0·94
No. 4 $E = 18 \times 10^6$ lb. per sq. in.	13·468	13·1	0·973
No. 5 $E = 18 \times 10^6$ lb. per sq. in.	10·014	6·7	0·668

AXIAL DEFLEXION OF A CRANK UNIT DUE TO PISTON LOAD.

Radial Load Tests on Crankshafts.—Mr. G. E. Windeler, M.I.Mech.E., has kindly supplied details of actual tests which enable the values of the empirical factors used in calculating the axial deflexion under piston loading to be determined. The pistons were loaded by means of compressed air, and the deflexion of the crank units was measured by dial indicators. Using the symbols employed in page 628, the essential dimensions were:—

a, 2.0 inches	A, 0.0121
b, 4.0 inches	B, 0.057
c, 2.125 inches	C, 0.00618

Comparison between Measured and Calculated Deflexion.—From equation (21), page 632, the axial deflexion for a load of 1000 pounds is 0.00396 inch; the actual test showed a deflexion of 0.0029 inch.

In the second tests, the values of a, b, and c remained as before, but the new value of A was 0.129, while B was 0.136 and C 0.00737, giving a calculated deflexion of 0.00066 inch per 1000 pounds, compared with a measured value of 0.000512 inch.

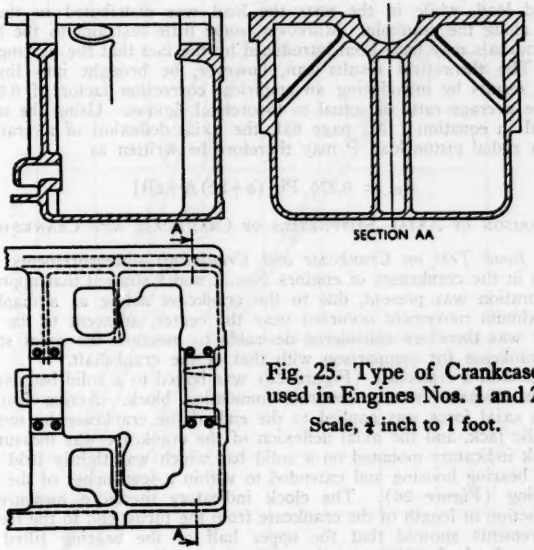


Fig. 25. Type of Crankcase used in Engines Nos. 1 and 2

Scale, ¾ inch to 1 foot.

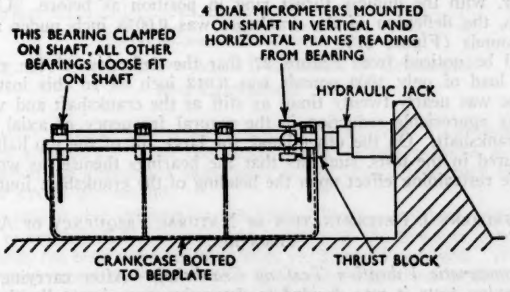

THIS BEARING CLAMPED ON SHAFT, ALL OTHER BEARINGS LOOSE FIT ON SHAFT
4 DIAL MICROMETERS MOUNTED ON SHAFT IN VERTICAL AND HORIZONTAL PLANES READING FROM BEARING
HYDRAULIC JACK
CRANKCASE BOLTED TO BEDPLATE
THRUST BLOCK

Fig. 26. Method of Loading Crankcase and Measuring its Deflexion

Scale, ¼ inch to 1 foot.

It should be pointed out that the calculated results are based on a concentrated load, while in the tests the load was distributed in the usual manner along the crankpin. Moreover, some little restraint to the bending of the journals may have been introduced by the fact that the bearings were fitted. The theoretical results can, however, be brought into line with the test results by introducing an empirical correction factor of 0.75, this being the average ratio of actual to theoretical figures. Using the notation employed in equation (13), page 631, the axial deflexion of a crank unit due to a radial piston load P may therefore be written as

$$\Delta_P = 0.375\ Pb[(a+2c)A+cB]$$

Comparison of Axial Stiffnesses of Crankcase and Crankshaft.

Axial Load Test on Crankcase and Crankshaft.—Measurements of the vibration in the crankcases of engines Nos. 1 and 2 showed that appreciable axial vibration was present, due to the crankcase acting as a diaphragm. The maximum movement occurred near the center, adjacent to the thrust ring. It was therefore considered desirable to measure the axial stiffness of the crankcase for comparison with that of the crankshaft.

For the test, a crankcase (Figure 25) was bolted to a solid bedplate at a convenient distance from a concrete foundation block. Acting from this block, an axial force was applied to the end of the crankcase by means of a hydraulic jack, and the axial deflexion of the crankcase was measured by four clock indicators mounted on a solid bar which was tightly held in the outboard bearing housing and extended to within a few inches of the thrust end bearing (Figure 26). The clock indicators therefore measured the total reduction in length of the crankcase from the thrust end to the free end.

Measurements showed that the upper half of the bearing tilted 0.014 inch with a load of 10,000 pounds, while the crankcase proper, at a point corresponding to the lower part of the thrust ring, moved only 0.0062 inch. The upper half-bearing was then removed and the load applied to the lower half only, with the dummy thrust ring in position as before. Under this condition, the deflexion of the crankcase was 0.0075 inch under a load of 10,000 pounds (Figure 27).

It will be noticed from Figure 27 that the deflexion of the crankshaft under a load of only 1000 pounds was 0.012 inch so, in this instance, the crankcase was nearly twenty times as stiff as the crankshaft and would not cause any appreciable reduction in the natural frequency of axial vibration of the crankshaft. On the other hand, the large tilt of the top half-bearing, as measured in the tests, suggests that the bearings themselves would have very little restraining effect upon the bending of the crankshaft journals.

Experimental Determination of Natural Frequency of Axial Vibration.

Electromagnetic Vibration Test on Crankshaft.—After carrying out the load-deflexion tests it was decided to determine experimentally the natural frequency of axial vibration, as a check upon the frequency calculated from the measured axial stiffness.

The crankshaft was carried vertically from a heavy steel beam arranged to form an arch across two large steel blocks, every precaution being taken to ensure rigidity of the supporting structure (Figure 28, Plate 2). To excite vibration in the crankshaft, an electromagnet was fixed beneath the free end, allowing an air gap of 0.05 inch from a small steel disk, which was fitted to the free end of the crankshaft in order to provide a suitable

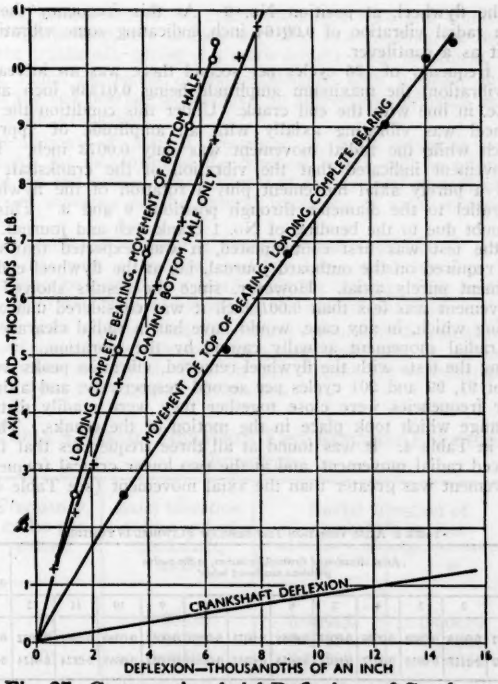


Fig. 27. Comparative Axial Deflexions of Crankcase and Crankshaft

magnetic circuit. Power was supplied to the magnet from a variable-speed motor-driven alternator. With this arrangement it was possible to apply to the crankshaft an alternating axial force of any chosen frequency between 25 and 1000 cycles per second.

Test Results.—A six-cylinder case-hardened steel crankshaft, similar in dimensions to crankshaft No. 6, was first tested with the flywheel in position, and afterwards the tests were repeated with the flywheel removed. With the flywheel, it was found that there were pronounced vibrations at frequencies of 90 and 126 cycles per second respectively.

To determine the nature of the oscillations, vertical vibration measurements were made at twelve equally spaced positions around the flywheel rim, position No. 12 falling in line with crank No. 1, the cranks being numbered from the free end. The results are given in Table 3 which also includes some of the radial vibration measurements of the flywheel, which were made as a check. Table 3 shows that at 90 cycles per second the maximum vertical vibration of 0.008 inch occurred at position No. 8, being in line with crank No. 2. The minimum vibration occurred at the opposite

side of the flywheel, at position No. 3. At this frequency there was a maximum radial vibration of 0.00164 inch, indicating some vibration of the crankshaft as a cantilever.

At the frequency of 126 cycles per second there was an increase in the vertical vibration, the maximum amplitude being 0.01238 inch at position No. 12, i.e. in line with the end crank. Under this condition the center of the flywheel was vibrating axially with an amplitude of approximately 0.0056 inch while the radial movement was only 0.0073 inch. The small radial movement indicated that the vibration of the crankshaft consisted mainly of a purely axial movement plus a rotation of the flywheel about a line parallel to the diameter through positions 9 and 3. This rotation was no doubt due to the bending of No. 1 crank web and journal.

When the test was first contemplated, it was expected that a bearing would be required on the outboard journal, i.e. at the flywheel end, to keep the movement purely axial. However, since the results showed that the radial movement was less than 0.001 inch it was considered unnecessary to fit a bearing which, in any case, would have had a radial clearance greater than the radial movement actually caused by the vibration.

Repeating the tests with the flywheel removed, vibration peaks were found to occur of 91, 99 and 201 cycles per second respectively, and although the two lower frequencies were close together they were readily distinguished by the change which took place in the motion of the cranks. The results are given in Table 4. It was found at all three frequencies that there was a pronounced radial movement, and at the two lower critical frequencies the radial movement was greater than the axial movement (see Table 4).

TABLE 3. AXIAL VIBRATION TEST RESULTS: FLYWHEEL IN POSITION

Frequency, cycles per second	Axial vibration of flywheel, in inches, at the twelve positions numbered below												Radial vibration of flywheel, inches	
	1	2	3	4	5	6	7	8	9	10	11	12	*a*	*b*
90	0·0031	0·0018	0·0009	0·0020	0·0047	0·0069	0·0073	0·0080	0·0051	0·0040	0·0007	0·0022	0·0016	0·0004
126	0·0102	0·0117	0·0058	0·0027	0·0022	0·0015	0·0022	0·0038	0·0055	0·0069	0·0124	0·0124	0·0007	0·0004

a, in line with crank No. 1; *b*, at right-angles to crank No. 1.

At 201 cycles per second the axial movement was 0.0040 inch, while the maximum radial movement was only 0.00255 inch so this frequency would approach the natural frequency of axial vibration. Since the crankshaft, without the flywheel, weighed 100 pounds and had an axial deflexion of 0.00309 per 1000-pound axial load, the estimated natural frequency would be 320 cycles per second compared with the measured frequency of 201 cycles per second. The addition of the flywheel, which also weighed 100 pounds, reduced the estimated natural frequency to 143 cycles per second, assuming the flywheel to be rigid; the measured natural frequency (Table 3) was 126 cycles per second. Since most of the flywheel mass was concentrated at the rim, and since the disk itself was relatively flexible axially, allowance had to be made for this flexibility. The axial deflexion of the crankshaft plus flywheel, measured from the rim, corresponded to a natural frequency of 132 cycles per second, only 6 per cent higher than the test value.

The following conclusions may be drawn from the tests:—

(1) The electromagnetic method appears to be very suitable for the study of crankshaft vibration problems and presents no serious difficulties in application.

(2) From comparison of the radial vibrations with and without the flywheel, the flywheel appears to have the effect of restricting the radial movement of the crankshaft, giving it a more definite axial movement.

(3) The radial movement of the crankshaft, with flywheel fitted, is so small that the end bearing would have no appreciable influence on the axial stiffness of the crankshaft.

(4) Calculations of natural frequency of axial vibration must make allowance for the movement of the flywheel, which will be affected by the bending of the end crank. The stiffness of the flywheel as a diaphragm has an important influence on the natural frequency of axial vibration of the crankshaft.

(5) The crankshaft without a flywheel is more subject to radial movement than that with a flywheel, and for large axial impulses the radial movement must be limited by a small-clearance bearing. The bearing would, therefore, have the effect of increasing the natural frequency.

(6) It is to be expected that the natural frequencies as calculated directly from tensile and compressive tests on a crankshaft will be higher than the value determined by actual vibration test because the calculation takes no account of radial movement of the masses.

Table 4. Axial Vibration Test Results: Flywheel Removed

Frequency, cycles per second	Axial vibration at free end of crankshaft, inches	Radial vibration of crankshaft, inches	
		a	*b*
91	0·00655	0·00965	0·00619
99	0·0040	0·00874	0·00801
201	0·0040	0·00218	0·00255

a, in line with crank No. 1; *b*, at right-angles to crank No. 1.

Analysis of Axial Vibration Records.

It has already been shown that the axial force set up in a crankshaft can be estimated from the measured axial vibration, but it is necessary to analyze the records in order to find the magnitude of the harmonics. The procedure described in textbooks is generally tedious and calls for extreme accuracy in drawing; recently, however, a short method of harmonic analysis has been developed primarily for electrical engineers, by Mr. S. Neville, B.Sc., a colleague of the author, and should prove of great value to engine designers. The records taken by the vibrometer give the resultant of the vibrations in individual cranks of the engine and the force calculated from the movement at the free end applies only to that end. The distribution of deflexion and force along the crankshaft can only be estimated when the stiffness of each crank is known for both radial and axial loading.

It can be shown that crank units having the same stiffness under *axial* load may differ in proportions as high as 10/1 in the deflexion under *radial* load; the axial force set up by a given radial load will therefore be in the same ratio. Thus, unless the stiffness ratio Δ_P/Δ_F is known, it is useless

to make calculations for multi-cylinder crankshafts by the vectorial method used in the study of torsional vibration problems. Unless further data are available, the author recommends that the axial forces should be calculated from the measured deflexion diagrams.

The most severe form of vibration occurring in the engine tests corresponds to a unidirectional pulsating force, such as is illustrated in the vibrograph record, Figure 17, at a speed of 1625 Rpm. The waveform is characterized by a series of half sine waves—indicating a rapid reversal of movement at the end of each oscillation—and can be represented by the Fourier series

$$\frac{4}{\pi}\left[\frac{1}{2} - \frac{\cos\theta}{3} - \frac{\cos 2\theta}{15} - \frac{\cos t 3\theta}{35}, \text{ etc.}\right]$$

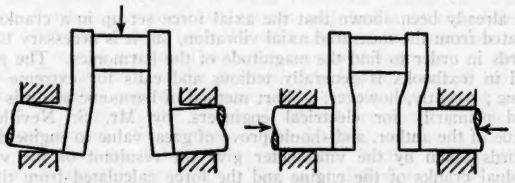
This series is made up of a constant value, together with odd and even negative harmonics. The second harmonic is of great importance, since its magnitude is one-fifth of the fundamental. The third harmonic is approximately one-eighth of the fundamental. Since the forces are proportional to the square of the frequency, the second and third harmonics are as important as the fundamental.

In certain cases the torsional vibration sets up a unidirectional axial vibration at twice the frequency of the torsional oscillation, but this is not a common characteristic. A second harmonic of axial vibration is also present, quite apart from that set up by the torsional oscillation, and the character of the resultant vibration depends entirely upon the magnitude and phase of the components due to pure axial motion and to the axial motion caused by the torsional oscillation.

Of the tests with which the author was concerned, one engine showed the double-frequency effect, but in another engine there was no sign of it in spite of the presence of pronounced torsional critical vibrations.

As a further aid to the interpretation of axial vibration records, the author finds it useful to study the cyclic deflexion of crank units by means of diagrams.

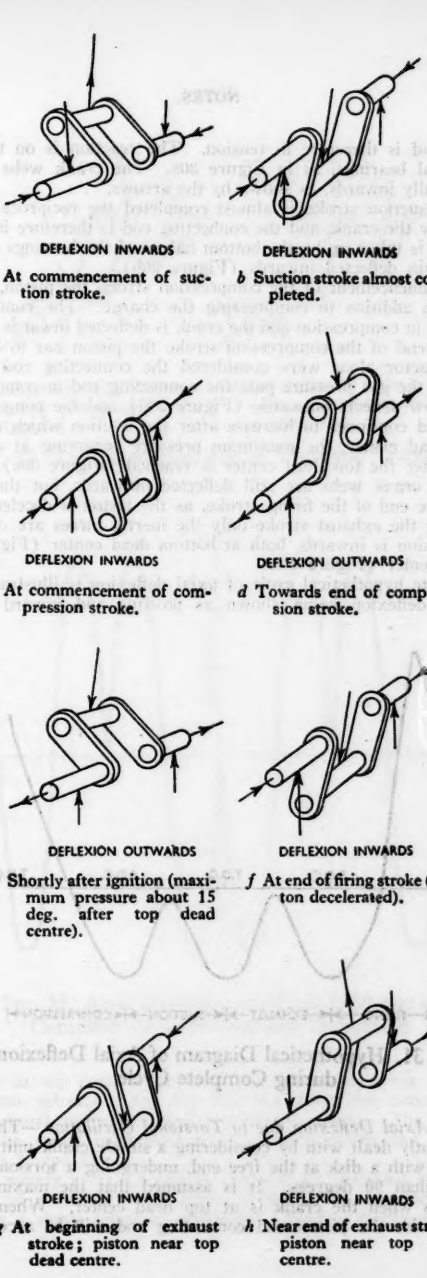
The Cyclic Change in the Axial Length of a Crankshaft.—As a crank unit is equivalent to an arch with semi-rigid supports, the crank webs tend to spread outwards when there is a radial load on the crankpin (Figure 29a) and so increase the axial length of the crank unit. Similarly, if the load is reversed, there will be a reduction in the axial length of the crank unit (Figure 29b).



a *b*

Fig. 29. Axial Deflexion of Crank Unit

Neglecting any axial deflexion due to twist in the webs, the cyclic axial deflexion of a single crank may be represented as in Figure 30a-b. At the commencement of the suction stroke the piston, etc., is accelerated and the



a At commencement of suction stroke.

b Suction stroke almost completed.

c At commencement of compression stroke.

d Towards end of compression stroke.

e Shortly after ignition (maximum pressure about 15 deg. after top dead centre).

f At end of firing stroke (piston decelerated).

g At beginning of exhaust stroke; piston near top dead centre.

h Near end of exhaust stroke; piston near top dead centre.

Fig. 30. Cyclic Deflexion of Crank Unit

connecting rod is therefore in tension. The reaction is on the top halves of the journal bearings, as in Figure 30a. The crank webs are therefore deflected axially inwards, as shown by the arrows.

When the suction stroke is almost completed the reciprocating parts are decelerated by the crank, and the connecting rod is therefore in compression. The reaction is taken up by the bottom halves of the bearings and the crank webs are again deflected inwards (Figure 30b).

At the commencement of the compression stroke the piston, etc., must be accelerated in addition to compressing the charge. The connecting rod is therefore still in compression and the crank is deflected inwards (Figure 30c). Towards the end of the compression stroke the piston has to be decelerated and if that factor alone were considered the connecting rod would be in tension. But the gas pressure puts the connecting rod in compression. The crank web now deflects outwards (Figure 30d) and the compression in the connecting rod continues to increase after the ignition which occurs a little before top dead center, the maximum pressure occurring at approximately 15 degrees after the top dead center is reached (Figure 30e). Under this condition the crank webs are still deflected outwards, but the deflexion is reversed at the end of the firing stroke, as the piston is decelerated (Figure 30f). During the exhaust stroke only the inertia forces are of importance, and the deflexion is inwards, both at bottom dead center (Figure 30g) and at top dead center (Figure 30h).

The complete hypothetical cycle of axial deflexion is illustrated in Figure 31, outward deflexion being shown as positive and inward deflexion as negative.

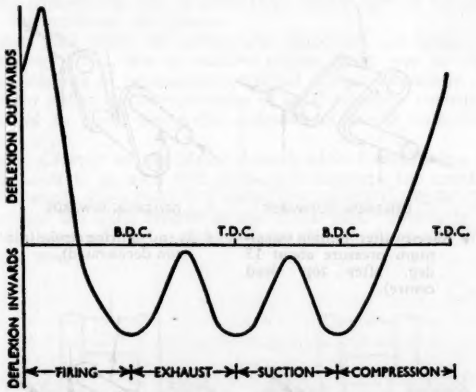
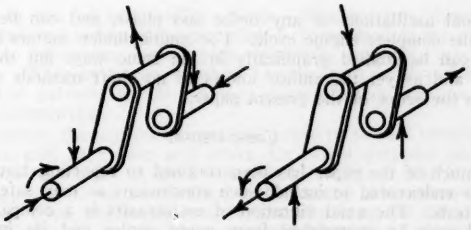


Fig. 31. Hypothetical Diagram of Axial Deflexions during Complete Cycle

The Cyclic Axial Deflexion due to Torsional Oscillation.—This is perhaps most conveniently dealt with by considering a simple crank unit fixed at one end and fitted with a disk at the free end, undergoing a torsional oscillation of not more than 90 degrees. It is assumed that the maximum angular velocity occurs when the crank is at top dead center. When the disk is rotating clockwise the piston and connecting rod will be accelerated from



Axial deflexion with disk accelerating piston and connecting rod.

Axial deflexion with disk decelerating piston and connecting rod.

Fig. 32. Axial Deflexion due to Torsional Vibration

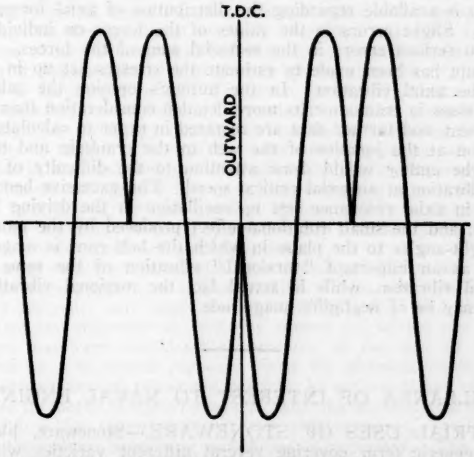


Fig. 33. Axial Deflexion of Crank Unit due to Complete Cycle of Torsional Oscillation

zero velocity at top dead center, and will take energy out of the system. As the angular velocity of the disk falls the piston and connecting rod will be decelerated and will give energy back to the system. The axial deflexion set up by the radial component of the inertia forces can be represented as in Figures 32 and 33.

Figure 33 merely gives the general shape of the deflexion cycle, and shows the number of reversals in direction of the axial movement, which take place during a single torsional oscillation. Similar diagrams can be drawn

for torsional oscillations of any order and phase, and can be constructed to cover the complete engine cycle. For multicylinder engines the resultant vibration can be studied graphically in the same way, but the process is laborious, and as yet the author knows of no short methods which might fall within the scope of the present paper.

Conclusion.

While much of the paper has been confined to reporting test results, the author has endeavored to include such conclusions as may safely be drawn from the tests. The axial vibration of crankshafts is a comparatively new study and may be approached from many angles and its importance is illustrated by the occasional occurrence of fractures (Figure 34, Plate 2) due to its influence. In the present paper the author has only been able to indicate some of the factors to be dealt with.

Regarding the calculation of the natural frequency of axial vibration, great accuracy is not to be expected until many more test results are available.

The effect of firing order has not been dealt with, since more data are necessary before the usual methods of analysis can be applied. It is impossible at the present stage to treat the problem vectorially as no definite information is available regarding the distribution of axial forces along the crankshaft. Slight errors in the values of the forces on individual cranks give rise to serious errors in the vectorial sum of the forces.

No attempt has been made to estimate the stresses set up in the crankshaft by the axial vibration. In the author's opinion the calculation of bending stresses in cranks merits more detailed consideration than it receives at the moment, and further data are required in order to calculate the stress concentration at the junction of the web to the crankpin and the journal.

Finally the author would draw attention to the difficulty of measuring torsional vibration at an axial critical speed. The excessive bending of the crankshaft in axial resonance sets up oscillation in the driving belt to the torsiograph, and the small rotational effect produced by the motion of the shaft at right-angles to the plane in which the belt runs is magnified until it appears as an important "torsional" vibration of the same frequency as the axial vibration, while in actual fact the torsional vibration at this frequency may be of negligible magnitude.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

INDUSTRIAL USES OF STONEWARE.—Stoneware, like plastics, is now a generic term covering several different varieties with distinct properties to meet different requirements. Industrial stoneware is available with compressive strength approximating that of grey cast iron and tensile properties exceeding those of the best cement. This brief article, describing certain ways in which stoneware is now utilized in various industries, is reprinted from Engineering (London), of May 1, 1942.

Stoneware is one of the three main classes of pottery, the other two being earthenware and porcelain. In view of the popular conception of pottery as essentially a fragile, delicate material, to regard stoneware as an engineering plant material may seem surprising. It is not generally appreciated, for example, that industrial stoneware is now available with a compressive strength comparable with that of grey cast iron and a tensile strength

superior to that of best quality cement. Neither is it always realized to what a growing extent stoneware plant is being used in the manufacture of chemicals, explosives, paper, pulp, dyestuffs, textiles, rayon, and food products. Other industries in which stoneware is employed include metal-plating and galvanizing, oil refining, storage-battery manufacture, tanning, and photo-engraving.

Even before the war, stoneware had replaced certain metals and metallic alloys, as well as rubber and wood, for many purposes and the present restrictions on the use of these materials has led to a still further extension of the applications of stoneware. Unlike many other plant materials, the ingredients from which stoneware is manufactured are all available in the United Kingdom. Among the more general types of stoneware plant are the following: tanks, jars, storage vessels, pumps, filters, coils, pipe-lines, cocks and valves, tile linings, digesters, and absorption, scrubbing and cooling towers.

Information concerning progress in ceramics is seldom found outside the ceramic journals and for this reason a short description of modern stoneware and its properties may be of interest. Owing to the fact that it was first applied in the purely chemical industries, the type of ware used for constructional purposes is often described as "chemical stoneware." This is a highly vitrified, dense ceramic material which, throughout its mass, is proof against the attacks of all acids and other corrosive agents, with the exception of hydrofluoric acid and hot, strong caustic alkalis. This property renders it independent of any surface glaze, although a glaze is generally applied to facilitate cleaning. The great variety of processes used in stoneware manufacture—including "throwing" on the potter's wheel, hand-moulding, dry pressing, casting and extruding—renders possible a great flexibility in design. Plant of the most intricate shapes can be fashioned in this plastic material and the dimensions of single pieces may range from a fraction of an inch to 6 feet or 7 feet; capacities of vessels range between a few ounces and several hundred gallons. Where the required design cannot be made in one piece, stoneware tiles are used for lining metal, concrete or wooden tanks and similar vessels.

Chemical stoneware is essentially an aluminum silicate and the composition of an average grade is as follows:—SiO_2, 73 per cent; Al_2O_3, 22 per cent; K_2O, 2 per cent; with small amounts of Fe_2O_3, CaO and Na_2O. The most important raw materials are ball clay, quartz and felspar and the proportions of these may vary considerably according to the type of stoneware body required for a particular purpose. From the physical standpoint, stoneware is an aggregation of refractory particles in various stages of fusion, bonded together by means of a vitrifying clay with the addition of a flux. Stoneware is fired at a high temperature—at least 1250 degrees C.—and constant supervision throughout processing and firing is necessary. The manufacture of this ware is a unique combination of craftsmanship and science, the leading makers in this country having been engaged in the industry for over a century.

The relationship between stoneware and the chemical industry in particular, may be traced back to the Middle Ages. Woodcuts of the Fifteenth Century show numerous alchemical distilling vessels made of this material and actual specimens are still in existence. In the early part of last century, the phenomenal expansion in the chemical industry would have been impossible but for the availability of corrosion-resisting stoneware. In more recent years, stoneware has had to face severe competition from new metals, alloys and synthetic materials but, despite this, it has not merely maintained its position but has found its way into many new fields, outside the purely chemical industries. Some of these have already been enumerated. One

effect of competition from other materials has been a great development in ceramic research, with consequent improvements in physical, chemical and electrical properties. In the stoneware field, particularly noteworthy progress has been made in improving compressive strength, tensile strength, bending and torsional resistances, and impact strength. Thermal conductivity has also been improved and, where necessary, the water absorption of unglazed ware has been eliminated.

Strictly speaking, it is no longer correct to speak of "stoneware" as though there were only one stoneware. Stoneware, like plastics, is now a generic term covering several different varieties with distinct properties to meet different requirements. One American maker claims to make twenty distinct stoneware bodies and several types are also available in this country. These include non-porous stoneware, the pore volume of which has been reduced almost to zero. At the same time, tensile strength is about 60 per cent greater than that of an average grade stoneware. The material is particularly useful in the food industries, as there is no possibility of minute traces of one batch being left behind in a vessel, tank or pipe to contaminate the next. It is also used for electrolytic cells, especially in the separation of precious metals, and in the manufacture of peroxides and perborates.

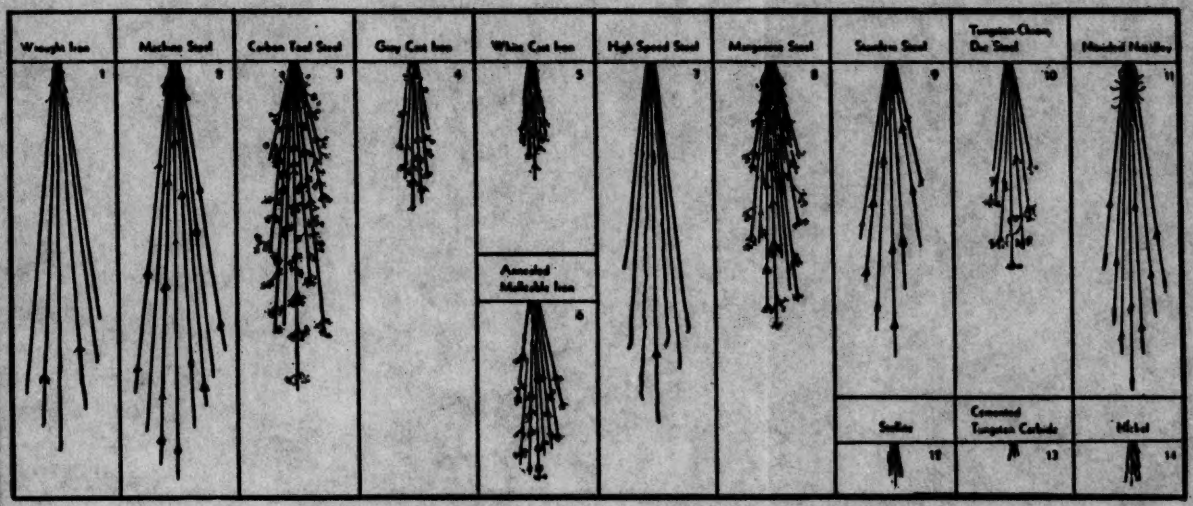
In heat-resisting stoneware, the thermal conductivity has been increased by 300 per cent and the thermal strength by 200 per cent in comparison with normal chemical stoneware. The pore volume, on the other hand, is much higher. It is used for plant in which heat is to be supplied or dissipated, such as condensing coils, pipe coolers, heat exchangers, kettles and evaporators. In the United States, it is claimed that equipment made from this ware is giving satisfactory service in molten lead and zinc baths at temperatures of 350 degrees to 450 degrees C. This ware is also more resistant to caustic alkalis. Thermal-shock resisting stoneware cannot be used at such high temperatures as heat-resisting stoneware, but it has a much higher degree of resistance to thermal shock than ordinary stoneware. Abrasion-resisting stoneware is a special grade containing corundum or fused alumina, and having a high resistance to abrasion and mechanical shock. It is used especially for the impellers of high-speed centrifugal pumps and exhaust fans. Because of the partial substitution of magnesium for aluminum silicate, alkali-resisting stoneware is slightly less resistant to acids, but has a much higher resistance to caustic alkalis. It also has a very high resistance to thermal shock, and is used for equipment subjected alternately to the action of acids and alkalis. The outstanding feature of white stoneware is its smooth white porcelain-like glaze applied to a body of greater mechanical strength than average stoneware. It is largely used in the manufacture of foodstuffs, beverages, drugs and cosmetics, where absolute cleanliness is essential.

The above examples may serve to indicate the versatility of modern stoneware. A universal or an ideal stoneware is out of the question, but by collaboration between the user and the manufacturer, it is generally possible to supply the type of ware which will suit a particular requirement. The principal objection which can be urged against the use of stoneware is that it is breakable. It must be remembered, however, that other materials also have their limitations. Iron rusts, cement is soluble in acids, wood is inflammable and rots, and the use of aluminum and its alloys is governed by numerous restrictions. Stoneware has an unequalled resistance to chemical corrosion and it affords great flexibility in design. It cannot rust, never needs painting and is easily cleaned. These qualities must be weighed in the balance against the one possible limitation; and, in numerous cases, this limitation can be overcome by skillful design and installation, with due regard to the fact that the tensile strength of stoneware is only a fraction

Material	Surface Appearance	Pellet Test after Sparking[1]	Magnet Test	Lathe Chip Test	Blowpipe Test
Gray Cast Iron	dull gray; evidence of sand mold		magnetic	short (⅛ in.) chips	moderate melting speed; quiet slag; quiet puddle
Austenitic Nickel Iron	gray; mold texture		non-magnetic		
Plain Carbon Steel	dark gray; may be rusty	dark, rounded, smooth surfaces	magnetic	long, often continuous	fast melting; quiet slag; puddle sparks
Nickel Steel	dark gray; surface condition varies		magnetic		
Chromium Steel	dark gray; surface condition varies	gray, frosty color; rounded particles	magnetic		
Tungsten Steel	dark gray; surface condition varies		magnetic		
Molybdenum Steel	dark gray; surface condition varies	(high carbon) hollow hemispheres	magnetic		
Vanadium Steel	dark gray; surface condition varies	black, truncated (hollow) shells	magnetic		
Straight-Chromium Stainless	dark gray; sometimes black-red rust		magnetic		
18-8 (or 25-12) Stainless	dark gray; dull to brilliant; usually "clean"		non-magnetic		
Inconel or Similar Alloys	dark gray; usually "clean"		non-magnetic		
Nickel	dark gray; smooth; sometimes green (oxide)		magnetic	cuts easily; smooth edges; can be continuous	melts slower than steel; less slag than Monel; quiet puddle
Monel	dark gray; smooth		slightly magnetic at room temp.	cuts easily; smooth edges; can be continuous	melts slower than steel; considerable quiet slag; quiet puddle
"Nickel Silver"	gray to yellow or yellow-green		non-magnetic	smooth, long, often continuous chips	moderate melting speed; fumes
Brass	yellow to green or brown		non-magnetic	smooth, long chips; more brittle than copper	moderate melting speed; gives off fumes; puddle like water
Tin Bronze	red to brown		non-magnetic	smooth, long chips; more brittle than copper	fast melting, some fumes; puddle like water
Aluminum Bronze	yellow-brown		non-magnetic		
Copper	smooth; red-brown to green (oxides)		non-magnetic	smooth, long easily cut chips	slow melting; very little slag; puddle bubbles
Aluminum	light gray to white; dull or brilliant		non-magnetic	smooth chips; saw edges where cut; chips can be continuous	very fast melting; quiet, black scum forms, quiet puddle
Magnesium Alloy	white; surface coating may be yellow-brown		non-magnetic	short, easily cut chips	
Lead	smooth, velvety; white to gray		non-magnetic	cut by knife; any shape chip	melts very fast; quiet slag, dull gray coating; quiet puddle
Lead-Silver Solder	smooth; dark gray; sometimes discolored		non-magnetic	harder than lead; any shape chip	slower melting than Pb or Pb-Sn
Lead-Tin Solder	smooth; white to gray; may be frosty		non-magnetic	harder than lead; any shape chip	faster melting than Pb-Ag; about same as lead

[1] The "pellet test" (Hildorf & McCollam, *The Iron Age*, Vol. 126, July 3, 1930, p. 1) is an adjunct to the spark test, and is especially useful for steels difficult to identify by the spark test. In the pellet test, grindings of the specimen are collected, passed through a 100 mesh sieve and the large pellets examined under an ordinary bench microscope. The pellets from various alloys vary characteristically in shape, texture and color.

Material	Chemical or Spot Tests				Spot Test Procedures
	1 Nitric Acid	2 "Iron Nail" Test	3 Ammonia Test	Others	
Gray Cast Iron	reacts; brown to black solution		red-brown	No. 4—No red color	*Test No. 1—Nitric Acid Test:* Place 1 or 2 drops of concentrated nitric acid on the clean metal surface. Observe any reaction for 1-2 min. Then dilute with 3-4 drops of water, and observe reaction. If solution turns green or blue, use it for the "iron nail" test.
Austenitic Nickel Iron	reacts slowly; brown to black solution		red-brown	No. 4—Intense reaction	
Plain Carbon Steel	reacts; brown solution		red-brown	No. 4—No red color	
Nickel Steel	reacts; brown to greenish-black soln.		red to purple	No. 4—Red color, intensity corresponds to amount of nickel	
Chromium Steel	reacts; brown to black solution		brown-green	No. 4—No red color in nickel-free steels	*Test No. 2—Iron Nail Test:* Rub a clean iron nail in the colored acid sol'n., in contact with the specimen. If the alloy contains copper, copper will be deposited on the nail or the metal surface.
Tungsten Steel	reacts slowly; brown soln., yellow sediment		red-brown		
Molybdenum Steel	reacts; brown to black solution		red-brown		*Test No. 3—Ammonia Test:* Dissolve a bit of the metal in nitric acid, or attack the surface with a few drops of acid (use aqua regia for stainless steel, Inconel, etc.). Dilute somewhat, carefully add ammonium hydroxide to the sol'n. until the latter is strongly alkaline. If copper or nickel is present, a pale blue precipitate will form, which on further ammonium addition becomes a dark blue sol'n.
Vanadium Steel	reacts; brown to black solution		red-brown		
Straight-Cr Stainless	no reaction		brown-green	No. 4—No red color	
18-8 (or 25-12) Stainless	no reaction		reddish-blue	No. 4—Intense red color; No. 6—Copper deposits on dilution	
Inconel or Similar Alloys	no reaction		blue-red	No. 4—Intense red color; No. 6—No copper is deposited	*Test No. 4—Paper Test for Nickel:* (Williams, *Ind. & Eng. Chem.*, Jan. 1942). One drop of acid mixture (10 ml. H_2SO_4, 10 ml. HNO_3, 10 ml. H_3PO_4, 10 g. citric acid, 25 ml. water) is placed on the metal for 15-30 sec., then absorbed in a paper test strip (filter paper dipped in a sol'n of 10 g. citric acid, 25 ml. H_2O, 10 ml. of 1% dimethyl glyoxime in isopropanol, and dried). When KOH is dropped on the paper, a red color will form if nickel is present. This test is free of color interference from iron or other elements.
Nickel	reacts slowly; pale green solution	negative	blue	No. 4—Intense red color	
Monel Metal	reacts; greenish-blue solution	copper plates out	dark blue	No. 4—Intense red color	
"Nickel Silver"	reacts; bluish-green solution	copper plates out	dark blue	No. 4—Red color varies in intensity with nickel content	*Test No. 5—Silver Nitrate Test for Tin:* Treat the clean metal surface with a few drops of 2½% silver nitrate sol'n. If tin is present, a black deposit with a white precipitate will form.
Brass	reacts vigorously; green solution	copper plates out	dark blue	No. 4—No red color in nickel-free alloys	
Tin Bronze	reacts vigorously; blue-green solution	copper plates out	dark blue		
Aluminum Bronze	reacts vigorously; blue-green solution	copper plates out	white ppt.; blue soln.		
Copper	reacts vigorously; blue-green solution	copper plates out	dark blue		
Aluminum	Soluble		white precipitate		*Test No. 6—Cupric Chloride Test:* This test serves primarily to distinguish 18-8 and similar stainless steels from low-iron nickel-base alloys like Inconel. Place 1 drop of a 10% cupric in conc. HCl sol'n. on the alloy for 2 min. Add 3-4 drops of water, slowly, then wash off. The high-iron alloys will develop a copper-colored spot on the metal surface.
Magnesium Alloy	Soluble		colorless		
Lead	Soluble		colorless	No. 5—No precipitate	
Lead-Silver Solder	Soluble		colorless	No. 5—No precipitate in absence of tin	
Lead-Tin Solder	Soluble		colorless	No. 5—Black deposit with white ppt.	



Metal	Volume of Stream	Length of Stream, in.	Color of Stream Near Wheel	Color of Streaks Near End	Quality of Spurts	Nature of Spurts
1. Wrought iron	Large	65	Straw	White	Very few	Forked
2. Machine steel	Large	70	White	White	Few	Forked
3. Carbon tool steel	Moderately large	55	White	White	Very many	Fine, repeating
4. Gray cast iron	Small	25	Red	Straw	Many	Fine, repeating
5. White cast iron	Very small	20	Red	Straw	Few	Fine, repeating
6. Annealed malleable iron	Moderate	30	Red	Straw	Many	Fine, repeating
7. High speed steel	Small	60	Red	Straw	Extremely few	Forked
8. Manganese steel	Moderately large	45	White	White	Many	Fine, repeating
9. Stainless steel	Moderate	50	Straw	White	Moderate	Forked
10. Tungsten-chromium die steel	Small	35	Red	Straw *	Many	Fine, repeating * Forked
11. Nitrided Nitralloy	Large (curved)	55	White	White	Moderate	
12. Stellite	Very small	10	Orange	Orange	None	
13. Cemented tungsten carbide	Extremely small	2	Light Orange	Light Orange	None	
14. Nickel	Very small **	10	Orange	Orange	None	
15. Copper, brass, aluminum	None				None	

† Figures obtained with 12 in. wheel on bench stand and are relative only. Actual length in each instance will vary with grinding wheel, pressure, etc.
* Blue-white spurts. ** Some wavy streaks.

of its compressive strength (7500 pounds per square inch against 116,800 pounds per square inch in the case of one particular grade). Within the proper limits and conditions of its employment, stoneware will last for centuries.

Stoneware plant, despite its apparent brittleness, can actually be worked with a hammer and chisel, if proper precautions are taken, and many fittings are regularly ground to precise measurements. Absorption towers, storage vessels and similar equipment may be erected out of doors and will give no trouble for decades, though constantly exposed to all kinds of weather. In certain paper and pulp mills, food factories and chemical works, stoneware pipe lines have been used to replace metal and have proved fully capable of withstanding considerable internal pressure, and have also eliminated the necessity for frequent replacement due to corrosion. The endurance of stoneware impellers in high-speed centrifugal pumps is another excellent example of the valuable mechanical properties of stoneware under appropriate conditions.

To detail all the ways in which stoneware is now being utilized in chemical engineering and other industries would require more space than can be devoted to it here, but perhaps sufficient indications have been given to induce those readers who have never employed this material to investigate its possibilities, especially now that so many other materials are in short supply.

RAPID SCRAP-IDENTIFICATION TESTS.—The Editors of Metals and Alloys compiled these reference tables of scrap-identification tests here reprinted from the August, 1942, issue of that publication. The "unknown" is usually known to be one of two materials, in which case the data for both materials in the Tables should be followed horizontally until a test is found in which the materials behave differently. If none such is found among the physical and chemical tests, the material may be identifiable by spark testing. These data were obtained from the International Nickel Co.; the Climax Molybdenum Corp.; the Linde Air Products Co.; Nassau Smelting and Refining Co., and the Norton Company.

Spark Tests for Classifying Metals.

Spark testing is an old and well-established method of classifying steels that has been used in steel mills and elsewhere for many years. In experienced hands, spark testing can be a very accurate—almost quantitative—method of identifying metals; even in less favorable circumstances the method can be a highly useful adjunct to salvage operations—particularly if a set of checking standards is prepared.

Spark tests should be made on a high-speed power grinder, with the specimen held so that the sparks fly off horizontally. The sparks should be examined against a dark background, and if possible, away from bright light.

The illustrations of typical spark streams given can only reveal their fundamental characteristics and serve as a very general guide. For best results, the specimen in question should always be spark-tested in comparison with actual samples of known composition.

One of the most useful aspects of spark testing is its ability to identify tungsten steels—otherwise very difficult without making a chemical analysis. Tungsten imparts a dull red color to the spark stream, near the wheel; it also shortens the stream and decreases the size of or eliminates the

carbon "burst." In 18-4-1 high speed steels, for example, the spark stream is long and thin, with dull red carrier lines. A 10 per cent W steel will have short curved orange spear points at the carrier ends. Further decrease in tungsten causes small white bursts to appear at the base of each spear-point—so delicate as to be unmistakably different from those of plain carbon steels.

Molybdenum-tungsten high speed steels have orange carrier lines, in contrast to the dull red lines of the high-tungsten steels. However, the orange carriers are also produced by other elements.

WELDING FUMES FROM ARC WELDING.—Dr. Gordon C. Harrold, Ph.D., of the Industrial Hygiene Laboratories, Medical Department, Chrysler Corporation, presented this paper at the Fifth Annual Tri-State Welding Conference, held at Pittsburgh, Pennsylvania, on April 24, 1942. His paper was subsequently published in the June, 1942, issue of the Welding Journal.

I. Introduction.

In facing the actuality of industrial hygienic shortcomings with regard to the exposures connected with arc welding, several things contribute to the obvious complexity of the situation. First, the quantitative determination of some welding gases, such as the several nitrogen oxides and ozone, is far more difficult than for many other substances. Second, the coating materials applied to welding rods are so diversified and so rapidly changed as to baffle almost any attempt to keep abreast of developments. Third, alloyed steels, such as nickel-chrome steel, or metals that have been coated, such as in galvanizing, are introducing new difficulties in connection with welding operations. As a result, the attending industrial hygienic situations are far from simple.

In general, the potential sources of injury from arc welding other than from trauma and thermal effects seemingly may be grouped in the following categories:

Oxygen deficiency.
Carbon monoxide.
Nitrous gases.
Ozone.
Metal fume from alloyed steel.
Metal fume from the metal electrodes.
Silica from welding rod coatings.
Harmful substances other than silica in rod coatings.
Injurious substances coated on to the metal being welded.
Radiations.

Possibly other gases such as cyanides theoretically produced in the electric arc.

From the fairly extensive first-hand experimental work and medical department experience with several hundred welders, we have reached a number of conclusions. These conclusions are developed first in regard to welding with bare rods, then coated rods and their by-products.

II. Bare Rod Welding.

Two hundred and fifty animals (rabbits and albino rats), including 87 controls, were exposed in five major series of experiments, together with

numerous casual and preliminary experiments, to the products of arc welding with bare, washed iron electrodes distributed over a period of 20 months. These animal exposures were attended by approximately 2400 analytical determinations for gases and fumes, embracing a minimum of 4 samples per hour during the daily welding cycle. The results of these animal exposures and accompanying analytical determinations lead to the following conclusions:

1. The quantity of "nitrous gases" (reported as nitrogen dioxide, NO_2) increased with every increase in voltage across the arc as utilized in welding. However, this statement must be considered in relation to the fact that with increased voltage the number of rods burnable per unit of time likewise increases.

2. The general average concentration of nitrogen dioxide (NO_2) in parts per million for increasing voltages across the arc were 29 ppm. at 27 v., 33 ppm. at 31 v., 40 ppm. at 33 v. and 70 ppm. at 44 v., under the uniform conditions of air change established in these experiments.

3. At a fixed voltage and after the period of actual welding, the "nitrous gases" remaining in the gassing chamber diminished as shown for 44 v.: 95 ppm. after welding, 73 ppm. 4 minutes after welding, 53 ppm. 10 minutes after welding, 46 ppm. 16 minutes after welding, at which time a new cycle of welding began.

4. The highest concentration of "nitrous gases" at any time averaged 97 ppm. for 10 samples with an extreme maximum for a single sample of 135, with a duplicate of 127.

5. Ozone (O_3) was found in amounts of from 10 to 32 ppm. within 1 inch of the arc, 1.5 to 9 ppm., 4 inches from the arc and from 0.2 to 1 ppm. in the center of the chamber, with only slight variations for different voltages, all samples being taken during the flaming of the arc.

6. As every welding day proceeded, after the first hour the concentrations for any given sampling period, such as 4 or 16 minutes after the cessation of welding, remained substantially the same. During the first hour of welding the concentrations increased.

7. Indicated, but not quantitatively determined, was the presence of nitric oxide (NO) in some unknown concentration along with the nitrogen dioxide (NO_2) in the "nitrous gas" mixture.

8. Determinations of "nitrous gases" through the measurement of nitrogen dioxide (NO_2) by means of the nitrite methods, which depend on the diazo reactions, unlike the nitrate methods, have not proved reliable in these arc-welding experiments.

9. Under the conditions of these investigations, distribution of the "nitrous gases" was substantially uniform throughout the gassing chamber. This statement does not apply to the distribution of ozone.

10. During animal exposures, percentages of oxygen and carbon dioxide were normal and there was detected no contamination by carbon monoxide, chlorine or other extraneous gases.

11. The quantity of particulate iron present as a fume and arising from the iron arc ranged from 35 to 395 mg. per cubic meter of air.

12. The quantity of iron fume present varied irregularly and sometimes inversely with voltage increases across the arc. It diminished after the cessation of welding.

13. No constant correlation was established between the respective quantities of iron fume, and "nitrous gases" within the atmosphere of the welding chamber.

14. The quantity of manganese present in the welding chamber atmosphere and derived from the burned welding rod varied from 0.2 to 3.3 mg. per cubic meter of air. The iron-manganese ratio in the air approximated the ratio

in the rod. Slight differences were encountered with some voltage variations which were attributed to alterations in the settling rates of the two types of particles.

15. No product of this experimental arc welding, regardless of voltage or duration of exposure through weeks, induced gross changes in experimental animals characteristic of the action of "nitrous gases."

16. The total exposure periods for animals, respectively, for four experiments were: at 27 v., 228 hours; 33 v., 205 hours; 44 v., 150 hours; 31 v., 120 hours. For these various voltages, the corresponding high average concentrations, respectively, were 32 ppm. for 50 hours, 47 ppm. for 45 hours, 84 ppm. for 33 hours, and 42 ppm. for 26 hours.

17. From all experiments, computations have been made as to the total number of hours of exposure of animals at various concentration levels. Thus in one experiment, where the average concentration was 70 ppm., the exposure was for 125 hours, not including the first hour. However, in this experiment, exposure was to 84 ppm. for 33 hours. With the exception of a portion of the first hour of every welding day, no exposure was to quantities of nitrous gases lower than 46 ppm.

18. In all experiments, the survivors exceeded 90 per cent of animals at the beginning of the experiment, not including those sacrificed within the experimental period or those dead from extraneous causes. In certain experiments, all animals survived save those selected for necropsy purposes.

19. When high atmospheric temperatures existed in the welding chamber, animals readily succumbed. The maintenance of such temperatures in the absence of any welding or welding gases led to high mortality from no other known cause than excess heat. When increasing temperatures without welding peaked at 106 degrees F. in an experimental chamber, when the average laboratory temperature approximated 72 degrees F., only 1 of 7 rabbits survived one 6½-hour exposure period. Ten rats, more resistant to heat, survived 2 days of exposure peaking for short periods at 106 degrees F. and were then subjected to increasing temperatures, whereupon all rats except one died. The highest temperature attained was 112 degrees F. for short periods only.

20. Autopsies were carried out on irregular numbers of animals during exposure periods, larger numbers immediately at the termination of the experiments and on others at irregular intervals after exposure. At all times, evidences of iron deposition were encountered as the sole constant manifestation of abnormality. Transportation of this ferrous material to the periphery of the lung and the tracheobronchial glands was the rule. In no animals was there encounted any gross pulmonary edema or edema along any portion of the respiratory tract.

21. X-ray examinations made after the termination of final exposure periods both on the intact animal and on the extirpated lungs yielded no positive results.

22. Limited exposures for human beings embraced 3 hours for two workers at an average concentration of 84 ppm. of nitrogen dioxide (NO_2), including a 40-minute period with exposure to 93 ppm. and a 5-minute period with an exposure of 103 ppm. No significant injury occurred.

23. In so far as indicated by these experiments, the threshold of harmful concentrations of "nitrous gases" lies above 70 ppm. for prolonged exposures.

24. In general, it has not proved possible to produce, under experimental conditions (which conditions exceed in severity those that attend usual industrial arc welding) sufficiently high concentrations of "nitrous gases" characteristically to affect any of fairly large numbers of animals, exposed daily over several weeks.

The above case for bare rods was then extended to deal with the more complex situations involving coated rods.

III. Coatings on Rods.

To meet these various purposes served by the rod coating, highly different substances enter into the coating mixture. Thus, to produce visible cloud about the arc, there may be present such ingredients as cotton lint, paper, powdered coke, etc. For the protecting slag layer immediately over the welding point, inorganic substances chiefly are used including borates, barium, calcium, aluminum and manganese compounds along with many others. In order to make up the deficiencies of the welding rod itself in relation to the metal to be welded, manganese, chromium, lead may enter into the coating itself. Lastly, it is necessary to bind these ingredients on to the rod, which calls for another class of materials, some of which in addition to being binders are gas generators. The number includes asphalt, resin, casein, ethyl silicate, sodium silicate, etc.

By way of practical objections to the use of heavily coated rods, welders have indicated that (1) it is difficult to observe the weld being made because of the overlying slag; (2) the cleaning of welds made with heavy coated rods is more difficult than the bare rod; (3) the increased visible clouds from the coated rods are disturbing and unwanted.

IV. Silica.

Large numbers of coated rods contain silica or silicate. The prime use of silicates is as a binder for other coating materials. Organic silicates are used for this same purpose, but this use is limited. The high temperatures (circa 6500 degrees F.) of the electric arc may convert the greater portion of combined silicon to silica. In the analysis of rod coating materials used in one of our series of experiments, the SiO_2 ranged between 17 per cent and 20 per cent. It has been computed that from the burning of a single 14-inch, 5/32-inch coated rod more than 150 mg. of fine particulate matter are introduced into the atmsophere. While this figure in itself is of little significance, it leads up to the statement that arc welding is or may be a dusty operation.

Dust from our welding showed 8 per cent silica. Experiments at Harvard have shown that this type of silica is not all free silica. General conclusions were that dust from coated rods containing silica or silicate are unlikely to constitute any practical threat of silicosis.

V. Summary of Coated Metallic Rod Experiments.

In addition to products created by the bare electrode were the new items, silica and titanium. No results suggest any injury from either of these agents.

Contrary to findings from bare rods, growth rate disturbances were encountered, chiefly in rats from which the case is unknown, though probably due to manganese.

Substantial quantities of methemoglobin were found in the blood of rats. In our previous animal work, no determinations were made for methemoglobin. Casual repetition of bare rod experiments established prompt methemoglobinemia in rats on the first day of exposure. A limited number of production welders were examined at a time when the NO_2 concentration in the work place was only 13 ppm. Methemoglobinemia was established. It is possible that, in the absence of other sources of methemoglobin, the presence of this entity in the blood of welders may indicate exposure to nitrous gases.

The quantity of nitrous gases produced by uniform electric power in case of the one coated rod investigated is distinctly less than in the case of bare rods at the same power.

Metallic fume in the gassing chamber is substantially increased when coated rods are employed. This is notably true of manganese fume in the coated rod used.

As in the case of bare rod experiments in which high quantities of nitrous gases were evolved, no evidence whatever accrued relative to pulmonary edema, severe respiratory tract inflammation or similar findings, traditionally associated with nitrous gas action.

Practical Applications and Implications.

1. In our experience and from our appraisal of the literature, it does not appear that the health of arc welders as a class may be set apart as especially different from metal workers in general. Although occasional records of deaths appear associated with some aspect of welding as the cause, we have no evidence of any prevailing diseases characteristic of the work of the arc welder (radiation effects excepted).

2. In our experimental studies, with the rate of rod burning much higher than in practical welding, and with welding confined to a comparatively small chamber of 1000 cubic feet, with at times only one air change per hour, we have not been able to find any suggestion of oxygen deficiency.

3. The much discussed point of ozone formation in the welding arc has in our experience resolved itself as follows: Much ozone (10-32 ppm.) is produced immediately in the arc, but quickly is diminished (1-9 ppm.) a few inches (4 inches) away from the arc and in the center of the chamber the residual is not more than 1 ppm. of ozone. Although this exceeds the somewhat accepted threshold of injury of 0.4 ppm., no injury from this gas has been determined by us.

4. Numerous tests for carbon monoxide have been made under various conditions—always without any significant results.

5. Other things being equal, low voltage and amperage are desirable in arc welding, since it appears to be true that low voltage is conducive to the low formation of nitrous gases.

6. Our best efforts to produce from arc welding sufficiently high concentrations of nitrous gases to induce injury to animals after long exposure were not successful. Practical arc welding in open areas seldom leads to higher concentrations of nitrous gases than 20 ppm.

7. It is our observation that concentrations of nitrous gases near the widely accepted threshold of 39 ppm. are quite harmless to laboratory animals after prolonged exposure and further that double this quantity for prolonged exposures and treble this quantity for limited exposures likewise are without discernible ill effects on animals. Human beings exposed to such quantities as 84 to 103 ppm. of nitrous oxides for a few hours (3) suffered no ill consequences.

8. We possess some evidence that methemoglobin is formed both in laboratory animals and human beings exposed to welding gases, but the quantity of this methemoglobin appears to be without injurious properties and further there is theoretical evidence that comparatively low percentages of methemoglobin in the blood may under some circumstances serve beneficiently.

9. Limited experimental observation is unfavorable to any belief that silicosis may be brought about from silica dust exposures incident to arc welding.

10. The heavy iron content of the air incident to animal exposure in welding has not been shown, from clinical, X-ray and autopsy observations, to produce any injury beyond pigmentation.

11. Apparently coated rods are to be favored over bare rods in so far as the production of nitrous gases is concerned.

12. Finally, in appraising arc welding as an industrial procedure and with particular reference to hygienic problems we at least are convinced that some over-emphasis in the past may have been placed on the significance of nitrous gases, carbon monoxide, ozone and iron fume. To the contrary, if there be practical hygienic problems related to arc welding (in addition to the obvious ultra-violet emanations), these may be expected in the little explored field of the constituents of coatings on rods, coatings on metals and special constituents of the metal welding rod or the metals being welded.

While the preceding material represents our laboratory's [1 2 3 4] conclusions on this subject, I wish now to briefly review the problem of metal arc welding in the light of the experimental work performed within the last 4 years. Extensive work has been done by Drinker [5 6] and his co-workers at Harvard. Tollman, MacQuiddy and LaTowsky and others at Nebraska [7 8] Britton and Walsh [9] at Northwestern and Von Haam [10] at Ohio State. While the Nebraska work was done with the carbon arc and carbon rods, all the other groups worked with the metallic rods. Except in minor details it is believed that their work is in full agreement with our own. Von Haam worked with three coated rods and produced much higher concentrations of fumes than will be found in practice except in small tanks. His conclusions were essentially to the effect that the one rod with high percentages of calcium fluoride showed much more damaging effects than the other coated rods worked with. A portion of the summary of Britton and Walsh's work, which included a survey of the literature and studies of a thousand welders, shows the following:

"The clinical portion of this report, based on a study of 286 welders of an average length of service of nine years, may be summarized as follows:

"1. While many hazards may exist as a result of the welding process, these can be and usually are controlled by the observance of ordinary precautions.

"2. Changes in the chest films after many years' exposure to welding fumes may occur. These changes are not accompanied by disability.

"3. In a comparatively large group of welders (about 1000) doing all types of welding, we have not observed any one disease entity to be outstanding in those disabled. Clinical syndromes in welders reported by others have not been observed.

"4. The disability frequency rate of welders has not been in excess of that found in a general group. The diseases found have otherwise not differed from those occurring in a general group of workers.

"5. There are hazards associated with any type of welding and troubles may develop as a result of exposure to these. However, when the hazards are considered and proper safeguards taken for their control, they should no longer offer a serious menace to health."

While the Nebraska group worked with treated carbon arcs, and while their quantities of fume and gases varied to some extent their conclusions in general bear out our own summations. They found that some high-intensity carbon arc ashes appear to cause proliferative reactions which injected intraperitoneally in rats. One statement is to the effect that rare metal salts, calcium phosphate, copper oxide and calcium fluoride appear inert. The rare metal salts mainly used are cerium oxide, lanthanum oxide, neodymium oxide and samarium oxide. It should be noted that copper

sulphate and other toxic salts do cause animal death when more than 10 mg. per day are present. Also it is well known that calcium fluoride is toxic. In the gases from the carbon arc from 72 to 1500 ppm. of NO_2 are found depending on ventilation, type of carbon arc, intensity of current and added constituents to the carbon.

While carbon monoxide has been stated by many to be a major factor, we have not found this to be the case with metal arcs, and their experiments show none in the carbon arc. At any rate the quantities stated in the literature would be harmless. Other gases which might have been considered, such as chlorine, phosphine and arsine, are relatively unimportant. While no ozone was found in the carbon arc, we have found definite quantities which have disappeared fairly rapidly due to interreaction with NO under the influence of ultra-violet light.

It should also be noted that the Nebraska group in experiments conducted on nitrous fumes indicate that harm is caused over a 7½-month period of time for quantities over 100 ppm. Our work with the metal arcs showed no harm over a considerable period of time when quantities upward of 70 ppm., but not over 100 ppm. were found in the atmosphere.

Very good agreement exists between most of the work done by Drinker and his associates and our work. In particular the studies by Tebbins of ventilation in arc welding when coated electrodes were involved confirms the statements above made and extends the material at hand in regard to the protective equipment needed. Their work indicated that the silica found was not in crystalline quartz but was probably in an amorphous form. Tebbins' work indicated also that ventilation probably is not needed except in confined spaces and that such ventilation, as is needed, becomes necessary mainly from the fume produced from coated rods rather than from the gases. Exceptional cases occur when low-boiling toxic material, such as lead or zinc, are found on the coated rods. He recommends the following ventilation rates when welding with coated electrodes:

Steel electrodes:	C. f. m.
5/32 in. or less	250
3/16 in.	400
1/4 in.	700
5/16 in.	1200
3/8 in.	1500
Alloy electrodes (fluoride coated):	
5/32 in. or less	250
Galvanized plate:	
5/32 in.	1000-1500

Tebbins also recommends suitable suction devices connected to local exhaust hoods placed within 8 to 9 inches of the arc with a minimum air velocity of about 125 feet per minute for smaller electrodes. This ventilation experience corresponds closely with what we have found necessary on the job, though it should be remembered that in large open areas it is seldom necessary to have any ventilation due to natural air movements.

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NEW TUBE-ICE MACHINE.—This article, describing Mr. B. F. Kubaugh's recent paper on the same subject delivered at a meeting of the American Society of Mechanical Engineers, is taken from the Transactions of the Institute of Marine Engineers, April, 1942. The abstract was earlier published in Modern Refrigeration, January, 1942.

A new method of producing hard, clear ice at once in small pieces of almost any desired size, has been evolved in the form of a tube-ice machine described in a paper recently presented by B. F. Kubaugh, at a meeting of the American Society of Mechanical Engineers, in Louisville, Ky. The new process is claimed to combine the advantages of both the plate and the can ice manufacturing methods. The tube-ice machine consists of a number of tubes of any diameter or length set on end vertically and enclosed in a shell, in the same manner as an ordinary vertical tubular open-ended condenser. This vessel is suitably supported over an inclined chute, consisting partly of solid metal and partly of screen or perforated metal. Below this chute is a water tank from which the water to be frozen is withdrawn by a pump and discharged into the top of the vessel, where it is distributed evenly to the several tubes. Each tube is fitted with a metal distributor which causes the water passing through its orifice to be projected against the inner surface of the tube, to which it clings in its descent. The circulation of the water provides the agitation which is required to produce clear ice. Below the vessel and above the chute is a rotating cutter, which not only nicks and breaks the ice as it descends, but also measures the length of the protruding pieces. Just below the lower tube head is a shear plate, attached to the tube plate by a ring, which forms a closed chamber. The latter serves as an auxiliary thawing chamber through which a very small quantity of warm water, at 85 degrees to 90 degrees F., is discharged at the desired moment to thaw the lower ends of the tubes, which protrude a short distance below the tube plate and are extended to and through the shear plate by means of copper ferrules to form the thawing chamber. The vessel itself is filled with the refrigerant—usually ammonia—to the required height, determined by a float-controlled liquid feed valve. Near the top of

the vessel is a horizontal transfer drum into which the refrigerant from the freezer is periodically discharged and from which it is expelled and returned to the freezer. An accumulator or liquid trap is attached to the transfer drum. At the beginning of the freezing operation a timing device, which controls all the functions, is started. On single-vessel installations, the timing gear also controls the water pump, starting and stopping it periodically but in the case of multiple-vessel installations, the water pump runs continuously, and the timing device merely controls the admission of the water alternately to the two or more vessels which comprise the unit. With the water in circulation, the timer opens the suction valve and the liquid feed, and the freezing process proceeds for a period predetermined for the suction or evaporating pressure employed, and for the thickness or kind of ice desired. At the termination of the freezing period, the timer shuts off the suction gas, the liquid feed and the water flow, and then admits high-pressure evacuating gas from the condenser, which displaces or removes the liquid refrigerant from the freezer into the transfer drum, and, at the same time, thaws the ice from the tubes. A few seconds after the beginning of the evacuation, the timing device starts the rotating cutter and admits warm water to the thawing chamber. Simultaneously with the removal of the liquid, the ice in the tubes drops by gravity on to the rotating cutter and is delivered to the chute in pieces of the desired size. All these operations take place in a matter of seconds; the whole process from the moment when the suction valve is closed to that when the last piece of ice has dropped into the chute takes barely a couple of minutes, regardless of the size of the freezing vessel. The discharged ice passes over the perforated portion of the chute into an ice storage tank. The general arrangement of a tube-ice machine of this kind is shown diagrammatically in Figure 1, while that of a multiple-vessel unit is shown in Figure 2, in which the shell-to-shell evacuating-system can be seen. The thickness of the ice made varies from a very thin shell to a solid cylinder, according to the duration of the freezing period, which is regulated by a simple speed control. The freezing period varies from 10 minutes for crushed shell ice at low evaporating pressure (around 15 lb./in.^{2}) to about 37 minutes for solid ice at high evaporating pressure (about 30 lb./in.^{2}). Where crushed ice is desired, the ice cylinders are relatively thin—about ½ inch—and short, whereas solid ice is produced in the form of cylinders of about 1¾ inches diameter by 2½ inches long. Other lengths are obtainable by merely fixing the measuring plate of the cutter at the required distance. Single freezer units have capacities varying from a fraction of 1 ton to 29 tons in "solid ice," and to 40 tons in crushed ice. A 10-ton single-vessel unit occupies a space of about 6 feet × 7 feet; a 40-ton single-vessel unit, 8 feet × 9 feet; a 65-ton two-vessel unit, 8 feet × 12 feet; and a 200-ton five-vessel unit, 9 feet × 30 feet or 18 feet × 20 feet. The tube-ice machine may be attached to an existing refrigerating plant, or it may be installed independently with its own "high side." compressor, condenser, receiver, etc. The operation of the machine is fully automatic and only requires occasional supervision, as nothing leaves the apparatus but the ice to be used. All small particles which chip off in the cutting process return to the pumping tank, contributing refrigerating effect to the circulating water. The timing mechanism consists of several cams of plastic material mounted on a horizontal shaft driven by a very small motor. Tube ice has many uses, and the machine may serve the dual purpose of cooling water only during a part of the day and freezing ice during the remainder by merely setting the suction valve to produce the required evaporating temperature for either purpose, or arranging two valves in parallel lines, each set for its respective pressure with the necessary stop valves for bringing each into service.

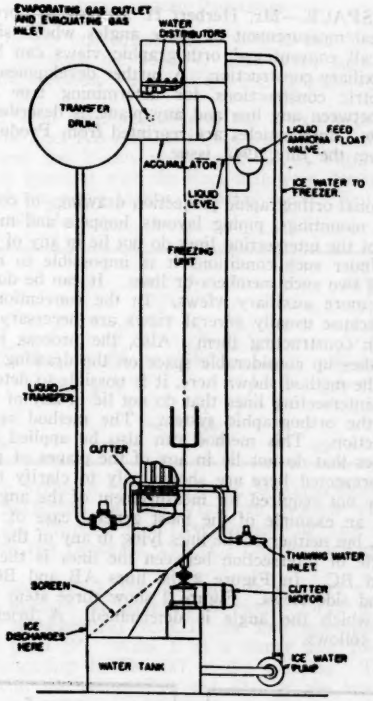

FIGURE 1.

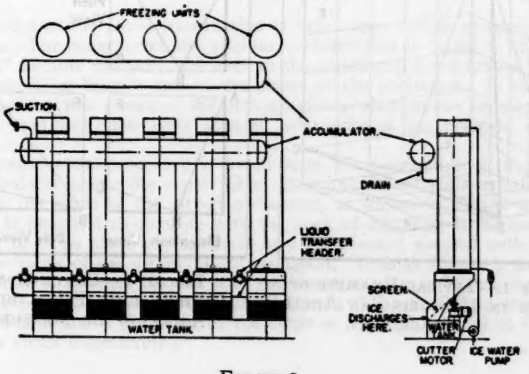

FIGURE 2.

ANGLES IN SPACE.—Mr. Herbert H. Stevens, Jr., presents a method by which graphical measurement of space angles whose sides are oblique to the planes of all conventional orthographic views can be accomplished with only one auxiliary construction. A further development of the method explaining geometric constructions for determining true angles between planes, and also between any line and any plane, is described in the article following this one. Both articles are reprinted from Product Engineering; the one below from the July, 1942, issue.

In the conventional orthographic projection drawings of constructions such as aircraft motor mountings, piping layouts, hoppers and many sheet metal structures, some of the intersecting lines do not lie in any of the three planes of projection. Under such conditions it is impossible to measure directly the angle between two such members or lines. It can be done only by constructing one or more auxiliary views. In the conventional method this is cumbersome because usually several views are necessary and errors are likely to occur in constructing them. Also, the process is tedious, time-consuming and takes up considerable space on the drawing paper.

By the use of the method shown here, it is possible to determine the angle between any two intersecting lines that do not lie in any of the three planes of projection in the orthographic system. The method requires only one auxiliary construction. This method can also be applied for determining the lengths of lines that do not lie in any of the planes of projection. The isometric views presented here are shown only to clarify the proof of the procedure and are not required for measurement of the angles or lines.

In Figure 1 is an example of the most general case of two lines intersecting at a point, but neither of the lines lying in any of the three projection planes. The angle of intersection between the lines is the angle made by the lines AB and BC. In Figure 2 the lines AB and BC are shown in plan, elevation and side views. Figure 3 show three steps in the auxiliary construction by which the angle is determined. A brief summarization of the procedure follows.

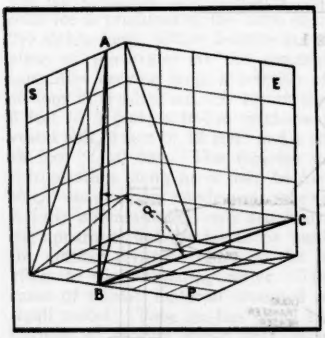

FIGURE 1.—TYPICAL EXAMPLE OF A PROBLEM IN MEASURING AN ANGLE IN SPACE.

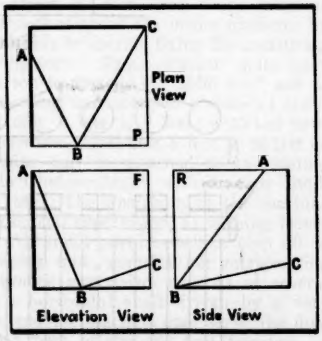

FIGURE 2.—CONVENTIONAL ORTHOGRAPHIC PROJECTIONS OF ANGLE TO BE MEASURED.

Summarized Procedure.

Draw a circle of any convenient radius with X and Y axes through its center point O. Through the point O draw lines parallel to the lines AB and BC in the elevation view of Figure 2. Figure 3(A) shows these lines.

From the point S′ at the intersection of the X-axis with the circle, draw lines parallel to the lines AB and BC in the side view of Figure 2. Where these lines intersect the Y-axis at A_s and C_s in Figure 3(B), project horizontal lines to intersect with the lines that were drawn parallel to the projected sides of the angle in the elevation view. Join these points of intersection, D_{AB} and D_{BC}.

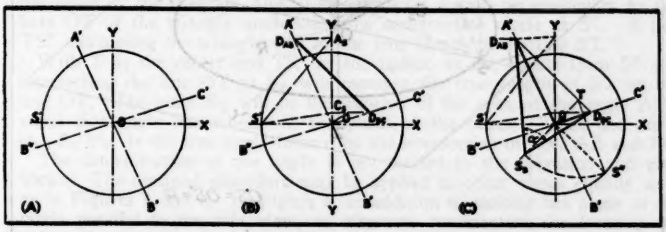

Figure 3.—Three Steps in the Graphical Measurement of the Angle Are: (A) Selection of the View to Which the Diagram Is Parallel, (B) Finding the Intersection of the Planes of the Angle and of the Diagram, and (C) Laying Out the True Angle.

Referring to Figure 3(C), the next step is to erect a perpendicular to D_{AB} D_{BC} passing through the center of the circle. Draw from the center of the circle a line perpendicular to OT, the line just drawn, intersecting the circle at the point S″. With T as a center and TS″ as the radius, swing an arc intersecting the line OT at the point S_B. The angle formed by joining S_B with the points D_{AB} and D_{BC} is the true angle between AB and BC, the legs of the angle.

Direction Diagram.

The proof of the above construction is based upon simple geometric relationships. For convenience the auxiliary construction is called a "direction diagram" because the principal lines in the construction are drawn parallel to corresponding lines in two of the planes of the projection. It should be noted that any two planes of the orthographic projection can be used. The elevation and side view were selected arbitrarily in the example outlined above.

The construction procedure is based upon the assumption of a point S perpendicularly above the center O of the circle at a height equal to the radius of the circle of the direction diagram, as shown in Figure 4. This point S is imagined to coincide with the apex of the angle to be measured, ABC or α. The plane of the circle may be assumed parallel with any of the three views of the orthographic projection. Thus in Figure 3 the plane of the circle is assumed parallel with the elevation view in Figure 2 and the lines A′B′ and B″C′ are drawn parallel with AB and BC in this view. (In Figures 5 and 6 the plane of the circle is assumed parallel to the side and plan views respectively.)

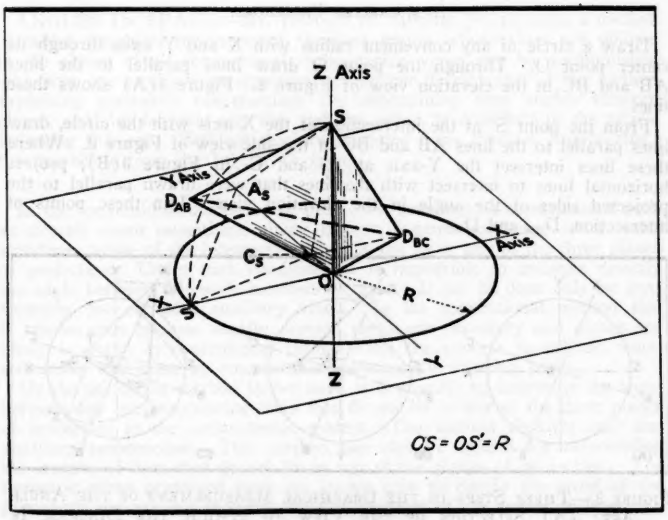

FIGURE 4.—PERSPECTIVE VIEW OF "DIRECTION DIAGRAM" SHOWING POINT S ABOVE THE PLANE OF THE DIAGRAM AND POSITION OF THE PLANE OF THE ANGLE TO BE MEASURED.

If two points are determined at which each of the two sides of the angle drawn from the point S in space intersect the plane of the paper, a line joining those points would locate the intersection of the plane of the angle ABC with the plane of the paper, or of the circle. With this intersection determined, it is a simple matter to rotate the plane of the angle about the line of intersection until the angle lies wholly in the plane of the paper where it can be measured.

These two points will lie on the lines A'B' and B''C'. The geometric relations involved in locating the points will be evident from the fact that if the plane of the paper is parallel to the front view, a plane through the point S and the Y-axis will be parallel to the side view. Lines parallel to the side view projections may be assumed to be in this plane. If this plane is rotated about the Y-axis the point S may be made to fall on S', because the distance OS perpendicular to the plane of the paper was defined as equal to the radius R of the construction circle. The directions of the projected sides of the angle in this plane now become parallel to projections in the side view of Figure 2. These lines are shown in Figure 3(B) as S'A_s and S'C_s.

These lines intersect the Y-axis and locate the distance from the X-axis at which the legs of the angles intersect the plane of the circle. This is evident from the isometric view in Figure 4. By drawing lines from these intersections parallel to the X-axis, the points D_{AB} and D_{BC} are located. These are the points at which the legs of the angle pierce the plane of the paper when the apex of the angle is at the point S above the plane of the

paper. The line D_{AB} D_{BC} is the intersection of the plane of the angle with the plane of the paper.

It is now necessary to rotate the plane $SD_{AB}D_{BC}$ in Figure 4 about the axis D_{AB} B_{BC} until the point S falls in the plane of the paper. From the geometry of the construction, the point S will rotate in a plane perpendicular to the paper and intersect the plane of the paper along the line OT in Figure 3(C).

The radius of rotation of point S will be the true length of the line ST, and also the hypotenuse of a right triangle whose other two sides are the distance between the center of the circle and the point S, and the distance between the center of the circle and the axis D_{AB} D_{BC}. This radius can be determined directly on the direction diagram by swinging this triangle into the plane of the diagram, that is, by drawing a line perpendicular to the base OT of the triangle intersecting the construction circle at S″. A line TS″ completing the triangle will be the true length of the line ST.

With T as the center and TS″ as the radius, an arc swung from S″ and intersecting the line OT at S_B will measure the true length of ST on the line OT. The point S_B will be the location of the apex of the angle ABC when that angle lies wholly in the plane of the paper. Thus, the angle D_{AB} S_B D_{BC} is the true angle formed by the intersection of lines AB and BC.

The determination of the angle is not limited to the elevation and plan views. The outlined procedure may be applied to other views equally well, as in Figures 5 and 6. In Figure 5, in addition to making the plane of the circle parallel to the side view, an alternate construction for locating the points D_{AB} and D_{BC} is shown. The plane through S and the X-axis, which is parallel to the plan view, is rotated so that S falls at S′ at the intersection of the Y-axis and the circle. The lines C_pS' and A_pS' are parallel to AB and BC in Figure 2, plan view.

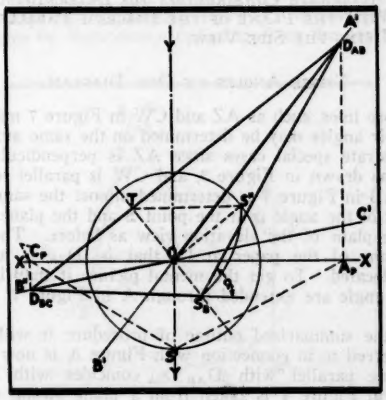

FIGURE 5.—AN ALTERNATE CONSTRUCTION FOR DETERMINING THE ANGLE α, STARTING WITH A PLANE PARALLEL TO THE SIDE VIEW AND USING THE PLAN VIEW PROJECTIONS TO LOCATE THE LINE OF INTERSECTION OF THE PLANES OF THE ANGLE AND OF THE DIAGRAM.

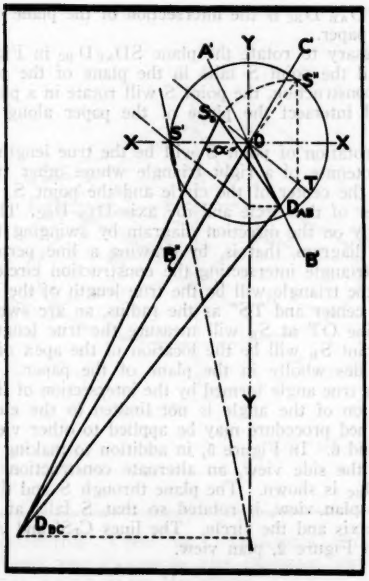

FIGURE 6.—AN ALTERNATE CONSTRUCTION FOR DETERMINING THE ANGLE α, STARTING WITH THE PLANE OF THE DIAGRAM PARALLEL TO THE PLAN VIEW AND USING THE SIDE VIEW.

THREE ANGLES ON ONE DIAGRAM.

When two more lines, such as AZ and CW in Figure 7 makes angles with AB and BC, their angles may be determined on the same auxiliary diagram. These lines illustrate special cases since AZ is perpendicular to the plane of the diagram as drawn in Figure 3, and CW is parallel to it.

The angle ZAB in Figure 7 is determined almost the same as before, assuming the apex of the angle is at the point S and the plane of the diagram is parallel to the plane of the elevation view as before. The line ZA must intersect the plane of the paper at O, that is, D_{ZA} is at the point O. D_{AB} is already located. To get the mental picture it may be assumed that the sides of the angle are extended through A in Figure 7, as indicated by the dotted lines.

In following the summarized outline of procedure it will be noted that the point T, referred to in connection with Figure 3, is now at the point O and that the line parallel with D_{AB} D_{ZA} coincides with the line. The desired angle β in Figure 7 is taken from a plane swung down from the space point S, as before.

Similarly, the angle BCW may be assumed to have its apex at the space point S. The line BC, prolonged through C, will intersect the plane of the diagram at the same point as in Figure 3. The line CW does not intersect the plane of the diagram because it is parallel to the plane. However, the

direction to the theoretical point at infinity is indicated as the dashed line through O parallel to the actual line CW, or its true projection in the elevation view.

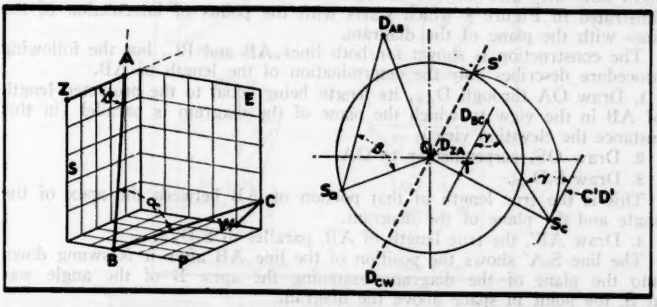

FIGURE 7.—CONSTRUCTION FOR DETERMINING TWO ADDITIONAL ANGLES ON THE SAME DIAGRAM USING POINTS PREVIOUSLY LOCATED.

In following the summarized outline of procedure for determining the angle, the points of intersection of the sides of the angle with the plane of the paper are assumed joined when a line is drawn through D_{BC} parallel to CW. The other steps are the same as outlined except that the angle is formed by joining S_e (corresponds to S_e in Figure 3) with D_{BC} and the imaginary point, DCW, the latter line being parallel to CW in Figure 7. This step may be omitted, however, because the desired angle may be formed also by lines shown radiating from the point D_{BC} as shown

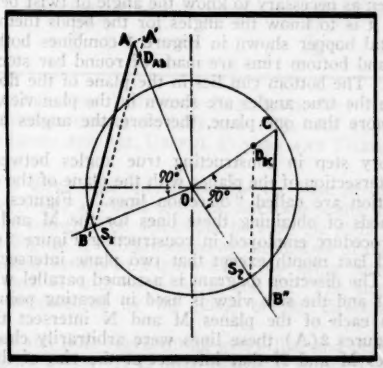

FIGURE 8.—CONSTRUCTION FOR DETERMINING TRUE LENGTHS OF LINES AB AND BC.

Length of Lines.

Since the lengths of lines such as those in the above example are usually required in addition to the angle between them, it is of further interest to show how they also may be determined on the direction diagram. This is illustrated in Figure 8 which starts with the points of intersection of the lines with the plane of the diagram.

The construction is shown for both lines AB and BC, but the following procedure describes only the determination of the length of AB.

1. Draw OA through D_{AB}, its length being equal to the projected length of AB in the view to which the plane of the diagram is parallel (in this instance the elevation view).
2. Draw OS_1 perpendicular to OA.
3. Draw S_1D_{AB}.

This is the true length of that portion of AB between the apex of the angle and the plane of the diagram.

4. Draw AB′, the true length of AB, parallel to S_1D_{AB}.

The line S_1A' shows the position of the line AB after it is swung down into the plane of the diagram, assuming the apex B of the angle was at S, the point in space above the diagram.

ANGLES BETWEEN PLANES.—This article by Mr. Herbert H. Stevens, Jr., complements the one on "Angles in Space" immediately above. This is reprinted from Product Engineering, August, 1942.

Problems in the determination of true angles between planes are encountered in the design of sheet metal structures, castings, bent tube and rod structures, and in piping layouts. In piping layouts compound angle measurement occurs when there are two or more bends not in the same plane. It is often as necessary to know the angle of twist between the planes of two bends as it is to know the angles for the bends themselves.

The sheet metal hopper shown in Figure 1 combines both of these problems. The top and bottom rims are made of round bar stock bent to shape before assembly. The bottom rim lies in the plane of the floor and is simple to measure since the true angles are shown in the plan view. The top rim, however, is in more than one plane, therefore, the angles are not as easily measured.

The preparatory step in constructing true angles between planes is to locate lines of intersection of the planes with the plane of the diagram. These lines of intersection are called "direction lines." Figures 2(A) and (B) shows two methods of obtaining these lines for the M and N sides of the hopper. The procedure employed in constructing Figure 2(A) is identical to that described last month except that two plane intersections are shown instead of one. The direction diagram is assumed parallel with the elevation view of Figure 1 and the side view is used in locating points at which two selected lines in each of the planes M and N intersect the plane of the diagram. In Figures 2(A) these lines were arbitrarily chosen as the three edges of the sides M and N that intersect at the rim of the hopper. One of these lines is common to both planes. The intersection of the three lines may be assumed to be at a space point S always perpendicularly above the center of the circle.

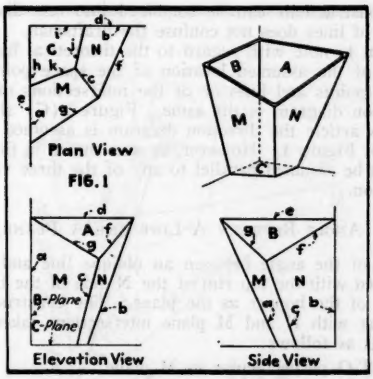

FIGURE 1.—PERSPECTIVE SKETCH OF HOPPER AND CONVENTIONAL ORTHOGRAPHIC PROJECTIONS FROM WHICH LINE DIRECTIONS ARE TRANSFERRED TO THE DIRECTION DIAGRAM.

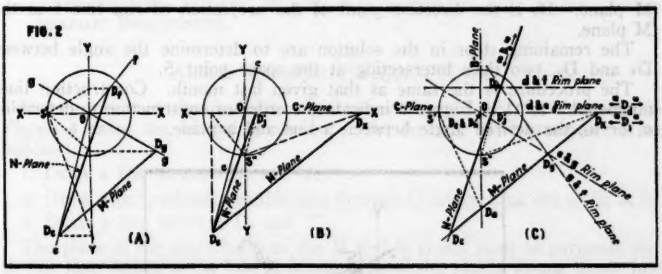

FIGURE 2.—TWO DIRECTION LINES FOR IMMEDIATE USE ARE PRODUCED BY DIFFERENT CONSTRUCTIONS AT (A) AND (B). AT (C) ARE DIRECTION POINTS AND LINES FOR ALL USEFUL PLANES AND THEIR INTERSECTIONS.

In Figure 2(B) three lines intersecting at the base of the hopper in the elevation and plan views are used to locate direction points in the two planes. This simplifies the construction somewhat because the plane C shown in the elevation view of Figure 1, is represented by the X-axis of the diagram. Still further simplification results in Figure 2(C) by using intersections of the hopper sides M and N with sides A, B and C. The sides A, B and C are respectively represented by the plane of the direction diagram, the Y and X axes.

Direction points and lines representing every line and plane of the hopper are shown in Figure 2(C). It is often desirable to use a separate diagram such as this and to trace off desired points and lines as required for determination of various angles. The direction diagram method is presented here in small segments and numerous diagrams but it will be obvious that

many of the constructions can be combined into one diagram as long as the multiplicity of lines does not confuse the draftsman.

It is important to note with regard to the theoretical basis of the method that regardless of the assumed location of the space point S, the location of the direction points and lines or of the intersections of lines and planes with the direction diagram is the same. Figure 2(C) shows this clearly. Throughout this article the direction diagram is assumed parallel with the elevation view of Figure 1. However, as was shown in the previous article, the diagram can be assumed parallel to any of the three views of an orthographic projection.

Angle Between A Line and A Plane.

Measurement of the angle between an oblique line and an oblique plane may be illustrated with the top rim of the N side of the hopper as the line and the M side of the hopper as the plane. The construction is shown in Figure 3 starting with N and M plane intersections taken from Figure 2. The procedure is as follows:

1. Draw line T_1O perpendicular to M plane.
2. Draw line OS′ perpendicular to T_1O.
3. Draw $S'T_1$.
4. Draw $S'D_p$ perpendicular to the line $S'T_1$.
5. Draw D_pD_f intersecting M-plane at D_q.

This line is the direction line of the plane of projection of rim line f in the M plane. D_q is the direction point of the projection of rim line f in the M plane.

The remaining steps in the solution are to determine the angle between D_f and D_q, two lines intersecting at the space point S.

The procedure is the same as that given last month. Construction lines numbered 6 to 11 in Figure 3 indicate the order of construction to determine α_1 or α_2, the desired angle between a line and a plane.

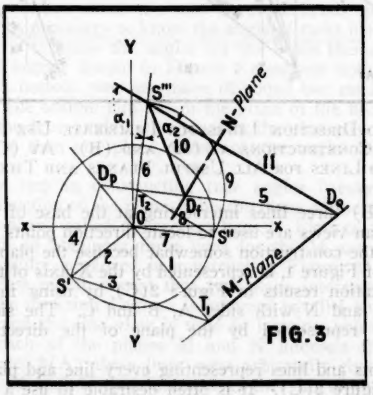

Figure 3.—Construction for the Angle Between Rim g and the N Side of the Hopper.

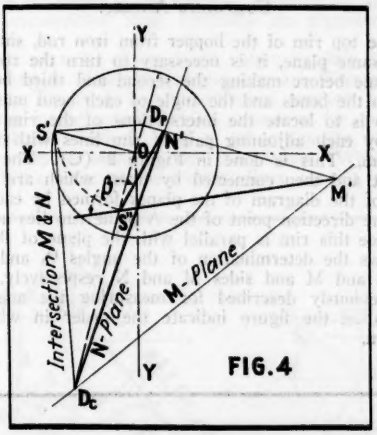

FIGURE 4.—CONSTRUCTION FOR THE MOST GENERAL CASE OF AN ANGLE BETWEEN TWO PLANES, M AND N, OBLIQUE TO ALL THREE ORTHOGRAPHIC PROJECTIONS.

ANGLES BETWEEN OBLIQUE PLANES.

Measurement of the angle between two oblique planes is illustrated in Figure 4 using the M and N planes of the hopper. The procedure is as follows:

1. Draw a line through D_c and O.
2. Draw a perpendicular to this line through O intersecting the circle at S'.
3. Draw a line between D_c and S'.

The plane of the angle between the M and N planes must be perpendicular to this line, which is the line of intersection of the planes swung down into the plane of the diagram. Therefore:

4. Draw S'D_p perpendicular to D_cS'.
5. Draw D_pD' perpendicular to D_cD_p.

This line is the intersection of the plane of the desired angle with the plane of the diagram. Since this plane passes through S and the desired angle has its apex at S, this angle must be swung down into the plane of the diagram as follows:

6. Lay off S'D_p on D_cD_p, thus determining point S".
7. Draw S"M' and S"N'.

The desired angle, β, is an outside obtuse angle for reasons which may be obvious from an examination of Figure 1. If the point S is assumed to be at the apex of the angle in the elevation view of Figure 1, that is at any point in the line of intersection of the M and N sides of the hopper, the M plane will extend away from the plane of the direction diagram while the N plane will extend toward this plane. The angle, therefore, must be measured between such extensions in the direction diagram.

Compound Angles.

In forming the top rim of the hopper from iron rod, since the four sides are not in the same plane, it is necessary to turn the rod on its axis in the bending frame before making the second and third bends. The angle of twist between the bends and the angle of each bend must be determined.

The first step is to locate the intersections of the rim lines and of the planes formed by each adjoining pair of rim lines with the plane of the direction diagram. This is done in Figure 2 (C). The direction points are located first and then connected by lines, which are the intersections with the plane of the diagram of the planes formed by each adjoining pair of rim lines. The direction point of the A plane rim lies at infinity parallel with X axis since this rim is parallel with the plane of the diagram.

Figure 5 shows the determination of the angles Θ_1 and Θ_2 between the rims of sides B and M and sides M and N respectively. The procedure follows that previously described for measuring the angle between two lines. Numbers on the figure indicate the order in which construction lines were drawn.

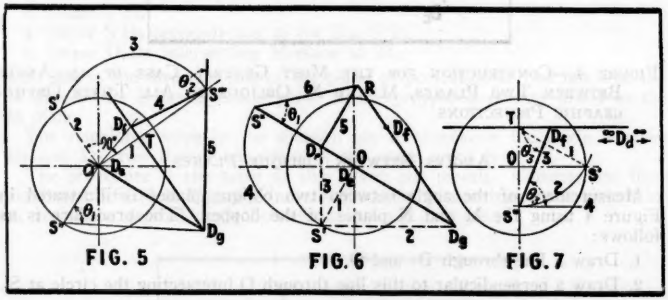

Figure 5. — Construction for Determining Angles Between Rims E and G and Between Rims G and F Give Degrees of Bend. Figure 2(C) Supplied Starting Lines. Figure 6.—Construction for Measuring the Twist or Side Angle Between Imaginary Planes Determined by Rims G and E and by Rims G and F. Figure 7.—The Angle Between Rims F and D Is Determined in This Construction.

The first angle of twist, ϕ_1, is determined in Figure 6. This is the angle between planes determined by the B and M rims and by the M and N rims and is, therefore, measured perpendicular to the M rim. The angle may be seen in the finished hopper rim by sighting along the rim of side M. Graphical determination of this twist angle follows the procedure outlined in this article for determining the angle between two planes. It is a special case because the plane of the B and M side rims passes through O. The order in which construction lines were drawn is indicated by the numbers on the figure.

The third angle, Θ_3, to be bent in the hopper rim is shown in Figure 7 using direction points and planes selected from Figure 2(C) as before. The construction corresponds with Figure 5 and the procedure is a special case

of the measurement of angles between lines. Numbers on the figures again give the order of construction lines, following the procedure outlined last month.

The second angle of twist, ϕ_2, is made before bending the rim for the corner of sides N and A. The construction, shown in Figure 8, corresponds closely with that in Figure 4, and the included numbers correspond with the numbered steps in the accompanying procedure for measuring angles between oblique planes. Figures 5 to 9 are ordinarily combined into a single diagram by anyone familiar with the method. When only the end results are sought the maze of construction lines are not confusing.

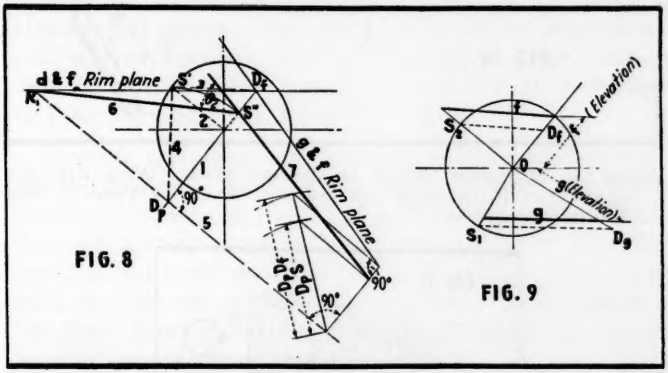

FIGURE 8.—CONSTRUCTION FOR DETERMINING THE TWIST OR SIDE ANGLE BETWEEN PLANES DETERMINED BY RIMS G AND F AND BY RIMS D AND F. A WELL-KNOWN MEANS IS USED TO LOCATE THE POINT OFF THE EDGE OF THE PAPER. FIGURE 9.—CONSTRUCTION FOR MEASURING THE LENGTH OF RIMS F AND G BETWEEN CENTERLINES OF ADJOINING SIDE RIMS.

HOPPER RIM DETAILS.

The lengths of rim sides between intersections of their center lines are easily determined on the direction diagram but further discussion of the determination of lengths between the bends is beyond the scope of this article, since it depends upon the pitch radius of the bending wheel. True lengths of d and e, the rims of sides A and B, are shown in Figure 1. Figure 9 gives true lengths of f and g, the rims of sides M and N, using the pocedure outlined last month. Figure 10 is a sketch of a simple bending wheel as used on table tops of many machines that are used to bend and form tubes and pipes.

Fabrication of the rim can be simplified by laying out a flat template like that in Figure 11 on which the three angles to be bent are shown to scale. The supplements of the angles Θ may be preferred by some. The template can be made on heavy paper, plywood or sheet metal. It need not be to full scale but should be large enough to lay the rim on to check the angles. The angle of twist between bends is indicated by the small circle diagrams. These show that the portion L_2 is twisted through an angle ϕ_1 and

L_3 through an angle ϕ_2. The direction of twist is indicated by whether the sloping leg of the angle is above or below the horizontal center line of the circle.

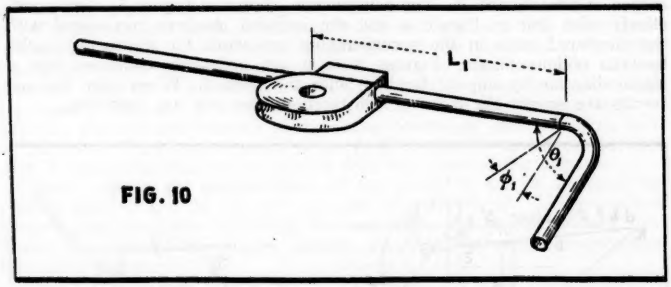

FIGURE 10.—SIMPLIFIED SKETCH OF BENDING WHEEL WITH RIM ROD CLAMPED IN PLACE READY FOR FORMING. ANGLE OF TWIST IN RIM ROD IS ALSO SHOWN.

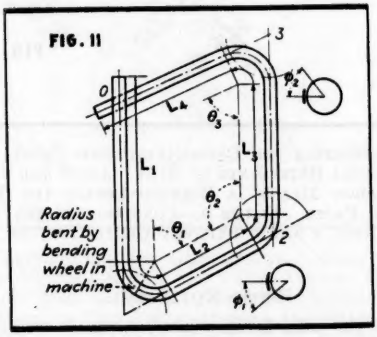

FIGURE 11.—COMPLETE RIM, SHOWS TRUE BENDING ANGLES AND SYMBOLS FOR TWIST ANGLES.

BOOK REVIEWS.

MARINE ENGINEERING, VOLUME ONE; EDITED BY HERBERT LEE SEWARD, PHB., M. E., PROFESSOR OF MECHANICAL AND MARINE ENGINEERING, YALE UNIVERSITY; PUBLISHED BY THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, 29 WEST 39TH ST., NEW YORK, N. Y., 1942.

Since the appearance of "Principles of Naval Architecture" and the decision of the Society of Naval Architects and Marine Engineers to publish a companion work on Marine Engineering, this book has been awaited by the engineering world. It is to be regretted that world conditions have prevented completion of the work and that only the first volume is ready for publication at this time. However, we can thank the Society for its decision to permit the one volume to appear.

This book is bound to become authoritative in its field. To permit covering the subject in the scope of two volumes, neither the history of marine engineering nor the principles of mechanical, heat and electrical engineering which comprise it are covered. The reader must seek elsewhere for those phases of the subject.

The true value of this book can be fully covered by stating its sponsorship which has been done and by a list of the Chapter headings and the author of each. These are:

Chapter I—Propelling Machinery, J. E. Burkhardt, Bethlehem Steel Co.

Chapter II—Power and Revolutions, J. E. Burkhardt.

Chapter III—Procedure in General Design, Benjamin Fox, Bethlehem Steel Co.

Chapter IV—Boilers, E. G. Bailey and T. B. Stillman, Babcock and Wilcox Co.

Chapter V—Reciprocating Steam Engines, J. C. Workman, American Bureau of Shipping.

Chapter VI—Steam Turbines, A. R. Smith and W. E. Blowney, General Electric Co.

Chapter VII—Diesel Engines, Louis R. Ford, Editor of "Motorboat and Diesel Boating."

Chapter VIII—Reduction Gears, C. R. Waller and A. Petersen, DeLaval Steam Turbine Co.

Chapter IX—Propellers and Shafting, Richard G. Tingey, Bethlehem Steel Co.

Chapter X—Materials and Metallurgical Engineerings, Paul Ffield, Bethlehem Steel Co.

HANDBOOK OF SHIP CALCULATIONS, CONSTRUCTION AND OPERATION By Charles H. Hughes. Published by McGraw Hill Book Co., Inc., New York and London, 1942.

Reviewed by Capt. N. W. Gokey, U. S. N.

By bringing out the third edition of his handbook at this time, Mr. Hughes has made a definite contribution to the war effort by providing an up-to-date reference book for naval architects, marine engineers, ship and engine draftsmen, deck officers, and others engaged in building and operating ships in the current program.

Although the new edition has been reduced from 740 to 558 pages, the contraction has been properly effected by careful condensation and the elimination of data that has no permanent value, such as fluctuating prices and costs for estimating purposes. Where condensation occurs, bibliographies are now provided to assist the user in obtaining more extensive data and information.

While the section headings remain as in the previous editions, rearrangement of subject matter in a more logical manner greatly improves and clarifies the presentation. Many sketches and diagrams have been redrawn to good purpose.

The greatest improvement which will be noted is the thorough modernization of the contents in line with the many developments and changes in trends which have occurred in hull and

machinery construction during the last 20 years, such as the advance of welding practice, diesel propulsion and air conditioning, to name only a few.

The new edition of the handbook should be of particular interest and value to the many newcomers in the field of ship construction and ship operation and can be highly recommended as an authoritative source of general and specific information on the subjects covered to which a good index affords easy and convenient reference.

INTRODUCTION TO NAVAL ARCHITECTURE, By John P. Comstock, Assistant Naval Architect, Newport News Shipbuilding and Dry Dock Co., Published by Simmons-Boardman Publishing Co., 30 Church St., New York, N. Y. Published September, 1942, Price $4.00.

This book is a text book on the subject of Naval Architecture. It is suitable for high school graduates who have not had college mathematics. Basically this book presents the course which the author uses in the instruction of hull drawing apprentices at the Newport News Shipbuilding and Dry Dock Co.

In the First Part of the book the subject of each chapter is applied to an "example" ship. There are numerous problems to be worked by the reader. The Second Part covers the design of a somewhat different ship through the various stages.

STATIC DEFENSE OF STRATEGIC OIL INSTALLATIONS, By B. Orchard Lisle, Scribner's Technical Book Dept., 597 Fifth Avenue, New York, N. Y. Price $1.00.

Preventative construction and protective concealment are the two phases of static defense dealt with by this manual, prepared by a member of the American Military Engineers. Designed in a format for convenient insertion in engineers' loose-leaf reference books, it contains 11,000 words of pertinent information based primarily on experience of oil men in war-torn Europe, in addition to contemporary American researches.

There are numerous photographic illustrations of both camouflaged and underground installations. There also are many drawings, charts and tables, showing bomb types, functions, and relative effectiveness—complimented by the degree of protection required against them.

In the first half of the text, the natures of both incendiary and demolition bombs are described, their effects analyzed, and precautions in construction and maintenance prior to air attack to reduce the hazard are given in the manual. The second half is devoted to the question of camouflage and total concealment.

OTHER BOOKS RECEIVED.

These books, although read with interest, are not reviewed because their nature is non-engineering.

AS THE TWIG IS BENT By Richard Welling, Published October 23, 1942, By G. P. Putnam's Sons, 2 West 45th St., New York City. Price $3.00.

LIFELINES OF VICTORY By Squadron Leader Murray Harris, Published By G. P. Putnam's Sons, New York. Price $2.00.

COMMANDO ATTACK, By Gordon Holmes, Published September, 1942, By G. P. Putnam's Sons, New York. Price $2.50.

ASSOCIATION NOTES.

Annual Meeting, 1942.

The annual meeting of the Society was held in Washington, D. C., on Tuesday, October 6, 1942.

Nominations for Officers for 1943.

The following were nominated for offices for the calendar year 1943.

For President:

Rear Admiral J. J. Broshek, U.S.N.,
Bureau of Ships, Navy Department.

For Secretary-Treasurer:

Captain J. E. Hamilton, U.S.N.
Bureau of Ships, Navy Department.
Lieut. Commander S. N. Pyne, U.S.N.,
Bureau of Ships, Navy Department.

For Member of Council:

Rear Admiral C. L. Brand, U.S.N.,
Bureau of Ships, Navy Department.
Rear Admiral E. L. Cochrane, U.S.N.,
Bureau of Ships, Navy Department.
Captain N. L. Rawlings, U.S.N.,
Bureau of Ships, Navy Department.
Captain L. F. Small, U.S.N.,
Bureau of Ships, Navy Department.
Rear Admiral E. W. Mills, U.S.N.,
Bureau of Ships, Navy Department.
Commander H. A. Carlisle, U.S.N.,
Bureau of Ships, Navy Department.

Mr. H. K. Beck, Dist. Sales Manager,
Worthington Pump & Machinery Corp.,
Washington, D. C.
Mr. P. N. Israel, President, Washington
Engineering Company, Washington, D. C.
Mr. A. F. E. Horn, Manager, Washington, D. C.,
Office, General Electric Company.
Mr. H. H. Rogge, Manager Washington, D. C., Office,
Westinghouse Electric and Manufacturing Company.
Mr. D. S. Brierley, Director, Division of Maintenance and
Repair, U. S. Maritime Commission, Washington, D. C.

Annual Banquet.

It was decided that unless war conditions change, the annual banquet of the Society will be omitted again in 1943.

Membership.

The following have joined the Society since the publication of the August, 1942, issue of the Journal:

Naval.

Angermann, Paul R., Lieut. Commander, U.S.N.R., Sales Engineer, Electric Switchboard Co., 42 Charlton St., New York, N. Y. Mail 130 Belford Ave., Rutherford, N. J.

Beach, Donald S., Ensign, U.S.N.R., Route 9, Box 161C, Seattle, Wash.

Brand, Charles L., Rear Admiral, U.S.N.

Brakel, Lewis H., Lieutenant, U.S.N.R., Navy Yard, Puget Sound, Wash. Mail 1936 Burwell St., Bremerton, Wash.

Doherty, Adrian W., Lieutenant, U.S.N.

Erbentraut, R. N., Sperry Gyroscope Co., Manhattan Bridge Plaza, Brooklyn, N. Y.

Gibson, A. R., Lieutenant, U.S.N.R.

Goldstein, Marton L., Lieut. Commander, U.S.N.R., 6731 Jeffery Ave., Chicago, Ill.

Hitchens, Gideon W., Commander, U.S.C.G.

Hoskins, Floyd E., Electrician, U.S.N.

Hume, J. W., Ensign, U.S.C.G., Coast Guard, St. Louis, Mo.

Huston, Robert J., Jr., Ensign, U.S.N.R., Mail 544 Winspear Ave., Buffalo, N. Y.

Kinsman, George H., Lieutenant, U.S.N.R., Navy Section Base, Portland, Me.

Lambine, Oscar M., Lieutenant, U.S.N.R.

McGrann, J. H., Lieutenant, U.S.N.R., R.O. in C. Bu. Y & D. Contracts, Kaneohu Bay N.A.S. Oahu, T. H.

Merser, Calvin S., Ensign, U.S.N.

Russe, F. W., Jr., Lieutenant, U.S.N.

Shimp, Robert P., Lieutenant, U.S.N.R., Office of Supervisor of Shipbuilding, Chicago, Ill.

Somerset, Donald E., Lieutenant, U.S.N.R.

Wheeler, George Oliver, Lieutenant, U.S.N.R., O. in C. Tech. Sec. Supervisor of Shipbuilding, Chicago, Ill. Mail 6731 Jeffery Ave., Chicago, Ill.

Wood, H. D., Ensign, U.S.N.R.

Wroblewski, Sigmund V., Ensign, U.S.N.R., 116 Middle St., Portsmouth, N. H.

Ziesing, Henry H., Jr., Ensign, U.S.N.R., Office of Supervisor of Shipbuilding, Philadelphia, Pa. Mail Wynnewood Park Apartments, Wynnewood, Pa.

Civil.

Currier, Charles H., Vice President and General Manager, Ross Heater & Mfg. Co., 1407-1411 West Ave., Buffalo, N. Y.

Gates, Walter H., Naval Architect, Standard Dredging Corporation, 80 Broad St., New York, N. Y. Mail 35 Maple St., Ridgefield Park, N. J.

Plesums, Michael J. P., Engineer, Van Der Horst Corp. of America, Olean, N. Y.

Santay, Joseph V., President, Combustion Engineering Company, 200 Madison Ave., New York, N. Y.

Associate.

Jester, Simeon Van T., Engineer, Bureau of Ships, Navy Dept. Mail Apt. 44, 2006 Columbia Road, N. W., Washington, D. C.

Walesa, Michael Aloysius, Metalsmith, Navy Yard, Mare Island. Mail, 1206 G St., Napa, Calif.

Podolski, Ivan, Lieut. Jugo Slavian Navy, Planning Engineering, Seattle-Tacoma Shipbuilding Corp, 2400 11th Ave., S. W., Seattle, Wash.

Woodward, Emery B., Asst. Electrical Engineer, Navy Yard, Charleston, S. C. Mail 1007 Montague Ave., North Charleston, S. C.

TRANSFERRED FROM CIVIL TO NAVAL.

Beeson, Kenneth H.
Gould, E. H.
Kurfess, W. F.

TRANSFERRED FROM ASSOCIATE TO NAVAL.

Flinn, Ernest A.